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HANDBOOK of

FLUID-FILLED, DEPTH/PRESSURE-COMPENSATING SYSTEMS FOR DEEP OCEAN APPLICATIONS

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DEEP OCEAN TECHNOLOGY PROGRAM

NAVAL SHIP RESEARCH and DEVELOPMENT CENTER

ANNAPOLIS LABORATORY

ANNAPOLIS, MARYLAND 21402

HANDBOOK OF
FLUID-FILLED, DEPTH/PRESSURE-COMPENSATING SYSTEMS
FOR DEEP OCEAN APPLICATIONS

By

Thomas H. Mehnert

April 1972

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ABSTRACT

Candidate approaches to depth/pressure-compensating devices and systems for fluid-filled deep submergence equipment are described and analyzed. Current design philosophies and considerations are discussed as well as the physical arrangement of compensating systems. A compensator design scheme and maintenance information have been included to provide guidelines and enumerate system design parameters and ramifications. It is planned that the handbook contents will be periodically revised and updated as "feedback" information becomes available.

PREFACE

The Deep Ocean Technology (DOT) "Handbook of Fluid-Filled, Depth/Pressure-Compensating Systems for Deep Ocean Applications" was prepared to provide guidelines for the selection, design, physical arrangement, and maintenance of compensating systems and devices. A design scheme for determining compensator volume allowances is presented and methods for fluid filling, fluid conditioning, and reconditioning are discussed.

The handbook is thus a guide, rather than a specification. It may be cited as authority for action, and will hopefully provoke deep submergence personnel to establish a thorough, yet realistic, approach in designing and maintaining reliable compensating systems. Users are encouraged to refer to other referenced DOT handbooks to obtain pertinent design information.

CONTENT AND ORGANIZATION

Chapter I relates the concept of depth-pressure compensation to environmental requirements and system design.

Chapter II is a description of compensating devices and systems as used in various deep submergence applications. Desirable design features, compensation schemes, and physical system arrangement are also discussed.

Chapter III includes compensating fluid considerations, a scheme for determining compensator volume allowances, and selected fluid data.

Chapter IV discusses fluid maintenance considerations and fill procedures for various equipment designs.

Chapter V acknowledges survey contributors and includes several complete deep submergence vehicle compensating system surveys. However, this survey information cannot be cited as authority for action.

REVISIONS, GROWTH, AND "USER COMMENT RETURN FORM"

The DOT "Handbook of Fluid-Filled, Depth/Pressure-Compensating Systems for Deep Ocean Applications" is designed to be periodically revised to include new data and considerations for fluid-filled system design and also for additional deep ocean applications. Responsibility for the maintenance and expansion of the handbook has been assigned, under the supervision of the Naval Ship Systems Command (SHIPS 03424), to the Naval Ship Research and Development Center, Annapolis, Maryland.

Revisions to the handbook will be effected by the use of page changes and additions. As the handbook is published in looseleaf form, revisions may easily be made.

Users, both commands and individuals within the Navy, and the nonmilitary marine community are encouraged to submit additional data, paragraphs, or chapters. Less extensive feedback - even mere indications that specified sections are judged to be too general - is useful and solicited. Feedback may be forwarded directly to:

Deep Ocean Technology Program
Naval Ship Research and Development Center
Annapolis, Maryland 21402

Material received will be carefully reviewed and coordinated prior to publication. A handy preaddressed user-comment return form is included for your convenience (following Bibliography).

ADMINISTRATIVE INFORMATION

This first edition of the handbook was compiled by the Naval Ship Research and Development Center, Annapolis, Maryland, as part of the Deep Ocean Technology Program, S4636, Task 14745, Work Unit 1-723-115-A "DOT Electrical and Hydraulic Systems for Deep Submergence Vehicles." The Program Manager was the Naval Ship Systems Command (SHIPS 03424), and the Naval Ship Engineering Center (SEC 6141) was the Technical Agent.

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This will acknowledge, with thanks, the cooperation of cognizant people in the naval laboratories, institutions, and industry who provided various background, design philosophy, and survey information for this effort.

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INTRODUCTION

The world's oceans are man's newest frontier for exploration, exploitation, and development. These activities involve a growing armory of vehicles and work systems to help man meet the challenges of hydrospace.

Whereas most operating machinery for a typical Fleet submarine is placed within the pressure hull, deep submergence vehicles (DSV's) employ external-to-hull machinery systems which must operate efficiently and reliably in the ambient ocean. Electrical control signals are transmitted from equipment in the pressure capsule to external systems by means of glass-to-metal seals and umbilical cabling. Vehicle designs often impose stringent technical constraints so that a DSV is not only weight limited, but space for mounting outboard hardware is at a premium. The extreme pressures, temperature, marine fouling, abrasive particulates, and corrosive action of sea water present difficult design problems.

Pressure, coupled with sea water, is decisively unforgiving. It ruthlessly penetrates and attacks the weak points of even the most sophisticated electrical, mechanical, or hydraulic components. Therefore, a design engineer must create a protective medium to shield sensitive hardware from the ocean. Shallow submergence design concepts frequently involve "hard shell" or encapsulation techniques. These approaches become impractical for deep ocean operational requirements, especially for large components and where rotating shaft penetrations into the seawater environment are required (see Background).

To operate in the deep ocean, the most feasible design approach is not to fight depth pressure, but to DESIGN FOR AND LIVE WITH PRESSURE. This can be accomplished by fluid-filling and pressure-compensating deep submergence systems and components.

A compensating system provides a protective fluid environment which is self-adjusting to be equal to or slightly greater in pressure than the ambient ocean environment. Various devices, such as a diaphragm, bellows, piston, or bladder, are often used to facilitate a moving interface between the compensating fluid and seawater.

Fluid-to-sea-water pressure differentials are then minimal. Structural housing and seal requirements are much less demanding than for a hard shell approach. The compensating device provides fluid volume compensation for changing physical conditions, both ambient and internal, such as occur during a DSV mission from the ocean surface to operational depth. A fluid, depth/pressure-compensated design concept allows the designer to meet critical equipment parameters - low weight, high reliability, compactness, and high efficiency.

BACKGROUND

The adaptation of machinery systems to perform useful work in the deep ocean has created problems which challenge the most innovative designer. Considering the obstacles, reasonable progress has been made in developing equipment which will meet performance requirements. Several approaches have been considered to adapt or shield system components for operation at depth in the ambient pressure, temperature, contaminants, and corrosive action of sea water.

SYSTEM ALTERNATIVES

SEA-WATER FLOODED

A machinery configuration which is open to sea water must be specifically designed to operate directly in the ocean environment. Advantages of this approach are that (1) a light machinery structure can be used; (2) a supply of machinery fluid is readily available; (3) dynamic seals are not required; and (4) good use is made of the ocean as a heat sink. Bearing configurations must be sea-water lubricated. Thus, this approach is limited to machinery systems where the corrosive, abrasive, poor lubricating, and conductive electrical properties of sea water can be tolerated.

ENCAPSULATION

Protection is achieved by completely surrounding components with a massive amount of material (such as epoxy resin). This technique is used by the electronics industry as a standard procedure for shielding components and circuitry from a moisture-laden environment. For applications in sea water at high pressure, a massive encapsulation which provides adequate environmental protection and electrical insulation will unfortunately be a good thermal insulator as well. Not only is heat transfer a problem, but the sealed component materials in contact with degradation products present a materials compatibility problem (see DOT Electrical Insulation Materials Handbook[9]).

HARD SHELL

This is a configuration whereby equipment is enclosed in a thick-walled, hermetically sealed case so that components operate in air or an inert gas at atmospheric pressure. The spherical or rib-stiffened cylindrical enclosure must be capable of sustaining test or "crush depth," which is nominally one

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[9]Superscripts refer to similarly numbered entries in the Bibliography, chapter VII.

and one-half times the operating depth of the vehicle. Since a hard shell must be opened for inspection and maintenance, a given design will involve flanges, bolts, and seals. This increases system size, air transportable weight, and submerged weight. In most cases, systems are already weight critical. A hard shell concept also involves additional expense in that high-strength steels or titanium must be used, with attendant pressure-cycling tests for certification.

Except for shallow submergence, this alternative is considered impractical where rotating shaft penetrations are required. Even if technologically feasible, frictional losses in "zero leakage" high-pressure shaft seals and associated thrust bearings would likely be prohibitive.

The penetration of a high-strength, thick-walled hard shell to bring in or take out electrical leads is an additional complexity. Moreover, heat removal is a significant problem. All heat-generating sources must not only be tied firmly to the interior of a hard shell, but must also be electrically insulated from it. As in problems of encapsulation, electronic components in this closed, in-air environment may incur materials compatibility problems. The previously cited DOT Electrical Insulation Materials Handbook deals in depth with the ramifications of several design approaches.

FLUID-FILLED, DEPTH/PRESSURE-COMPENSATED SYSTEM

To achieve corrosion and electrical protection and also lubricate machinery components, a system may be filled with a suitable fluid* and provided with a depth/pressure-compensating device. Ambient external pressure is transmitted into the fluid through a flexible interface such as a bladder, diaphragm, bellows, or other devices. Differential pressures across enclosure penetrations and the associated seals are minimal. Structural problems and enclosure weight are reduced. Static and dynamic sealing concepts take on a unique design requirement - that of low-pressure differential seals functioning in a cyclic high-pressure fluid environment.

Compensating fluid heat transfer characteristics are better than those of encapsulating materials, or of air, as in the case of the hard shell concept. System reliability may be enhanced by selecting fluids which have qualities of inhibiting sea-water corrosion. Thin-walled enclosures may be readily disassembled for equipment inspection, trouble-shooting, or maintenance. To adapt and protect machinery systems for operation in the deep ocean, a fluid-filled, depth/pressure-compensated approach has

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*The term "fluid," as used in this handbook, will include all candidate compensating fluids with the exception of fresh water and sea water.

been preferred. This handbook will then expand upon various considerations in the design and maintenance of depth/pressure-compensating systems. In accordance with original guidelines for this task, the compensation of outboard batteries is not specifically addressed.

CHAPTER I

DESIGN RATIONALE AND COMPENSATING SYSTEM REQUIREMENTS

1.1 INTRODUCTION

It is evident that there has been significant historical interest in deeper operating vehicles, whether they be for scientific and research programs or for military purposes. During the development of a DSV design, a multitude of parameters must be manipulated to satisfy the specific needs of the desired vehicle.

On the one hand, an engineer will strive for equipment designs which provide high power-to-weight ratios, high reliability and efficiency, light weight, long life, and even low noise. But basic to all of these considerations is that both power and auxiliary systems are located external to the personnel capsule, directly in the ambient ocean environment. Usually the reasoning behind this design approach is to minimize the size of the pressure hull and to eliminate any mechanical hull penetrations. Mounted outboard of the pressure hull, equipment enclosures also tend to be somewhat self-buoyant. The surrounding sea water affords a virtually infinite heat sink. For a manned DSV, only electrical control and display signals will penetrate the spherical or cylindrical personnel capsule. Internal space for the crew and essential equipment is conserved. Moreover, external systems and components are removable without compromise to pressure hull integrity.

With the hard shell concept, components are not subjected to pressure cycling. Often it has been necessary to hard shell electronic circuitry or other sensitive components in order to obtain good stability and reliability. Depth/pressure-compensated hardware must be adaptable and fully compatible with a cyclic, high-pressure-fluid environment for all modes of operation. Due to the complexity of a given compensating system and the associated fill, drain, and cleaning procedures, a hard shell enclosure may be much simpler to open and close. However, the designer must also consider the weight and space penalty, frequency of maintenance, heat dissipation, etc, for the hard shell approach.

After considering various system design alternatives, generally the most attractive approach is to fluid fill and depth pressure compensate equipment enclosures. Compensating systems, as used in deep

ocean applications, provide a protective fluid-filled environment which is self-adjusting to be equal to or slightly greater in pressure than the ambient ocean environment. Bascially, this is the scope of this handbook.

1.2 FLUID COMPENSATION RATIONALE

1.2.1 In simplest form, the compensating fluid which fills an equipment enclosure is balanced or equalized to all variations of ambient sea water or atmospheric pressure, thus reducing the differential pressure to approximately zero. A flexible moving interface, such as a bladder or diaphragm, is used to communicate external pressure into the compensating fluid and allows system fluid to expand or contract.

1.2.2 A further design consideration is to maintain a slight positive bias or Δp above ambient pressure on the compensating fluid for applications where dynamic seals are employed. Should leakage occur, the rationale is to encourage outward leakage and minimize inward leakage.

1.2.3 Outboard equipment can function at depth in a protective, electrically insulated, lubricating-fluid environment. Conventional bearings and gears may be used. The compensating system must provide for internal changes in fluid volume which are dependent upon:

- Thermal expansion or contraction of the fluid.
- Fluid compressibility.
- Leakage out of and into the system.
- Fluid gassing - from heat produced by the mechanical action of gears and bearings, electrical arcing (contactors and commutation), and electrical hot spots.

1.2.4 Since equipment enclosures or housings must withstand only a minimal differential pressure, structural and sealing problems are minimized.

1.2.5 System low points should contain fill/drain ports while high points should contain ports for venting or filling, depending upon the most feasible fluid-fill procedure. If vacuum filling of components is to be done, the compensating system must make the proper provisions.

1.2.6 The choice of a sea-water emulsifying or non-emulsifying fluid will depend upon system requirements and design philosophy (refer to section 3.3 for a discussion of various considerations).

1.2.7 Sea-water leakage detectors may be installed in low points of the system to provide a warning to operating personnel of sea-water intrusion.

1.2.8 Lightly spring-loaded check valves may be used to relieve internal overpressure in the compensating system. Should the volume of the fluid in the compensating device become depleted, check valves may admit sea water as the emergency compensating fluid to facilitate a "fail-sick" operating condition.

1.2.9 Trapped fluid in hydraulic circuits or otherwise uncompensated fluid volumes are relieved during ascent and descent modes to preclude damage and malfunction of seals and components.

1.2.10 In fluid power applications, the compensator may function as a hydraulic accumulator, absorbing pulsations and cushioning operating shock.

1.3 ENVIRONMENTAL REQUIREMENTS

1.3.1 Pressure Cycling

During a DSV mission, from surface to operational depth, compensating systems must respond to cyclic pressurization with acceptable hysteresis and maintain the integrity of fluid-immersed hardware under all operating conditions. Component materials must not incur stress fatigue or permanent set from pressure-cycling.

1.3.2 Temperature Extremes

The following ranges of temperature are given as examples of environmental and operating extremes for compensating systems. The particular DSV mission requirements and system operating characteristics will determine what temperatures are realistic for calculating fluid volume allowances.

Nonoperating Range (Storage and Transportation)	-40° to 140° F*

* Abbreviations used in this text are from the GPO Style Manual, 1967, unless otherwise noted.

Operating Range (Surface and Submerged)	28° to 190° F
Sea-water Ambient	28° to 90° F

For example, the very high operating temperature cited above would be representative of an electric drive motor operating under full load at the sea surface with poor cooling conditions.

1.3.3 Pressure Extremes

Vacuum (fluid fill procedure) or Subatmospheric (air transportable)	To test depth (usually 1.5 x operational depth)

1.3.4 Materials Compatibility/Corrosion

A DSV may spend much more time at the surface, or out of the water, than submerged at depth; therefore, surface environment must also be considered. Corrosion mechanisms in moist, salt-laden air are similar to those in sea water, with perhaps even more effect due to the high oxygen content.

Component materials should be selected for their compatibility with one another when exposed to the sea-water environment and/or system fluid. Seal materials must not only be suitable for high ambient pressures but should not suffer excessive swelling, compression, or permanent deformation. Unless totally immersed in a suitably inhibited compensating fluid, metallic materials should be corrosion resistant in sea water and compatible with regard to electrolytic action. Within a compensated system, corrosion will increase wear directly at contacting metal surfaces and indirectly by generating hard oxide particles which will act as abrasives. As a safety and reliability goal, hardware which is normally fluid-filled and pressure-compensated should not readily deteriorate or become inoperative when flooded with sea water for a short time.

While metallic materials such as type 316 stainless steel, 6061-T6 aluminum, and Monel have commonly been used in this application, the use of titanium alloys should provide the basis for even greater capabilities and reliability.

> Titanium is useful in a marine environment primarily because it is corrosion resistant, and has an excellent strength-to-weight ratio (up to 1.7 times that of steels). In addition to resisting general corrosion, titanium has good resistance to crevice corrosion, and the highest resistance to corrosion fatigue of any structural metal. It resists impingement attack and cavitation. While the cost per pound of titanium is high, the U.S. Navy is interested in it because its high strength-to-weight ratio makes its actual application cost less than that of Hastelloy C, and competitive with monel. Moreover, the cost per pound of titanium is decreasing continually with increasing commercial use, so that, even on a first-cost basis, titanium is becoming more nearly competitive with such metals as stainless steels. The cost advantage is further increased by an anticipated longer life of titanium parts in many applications.[4]

Extensive data have been published on the selection of corrosion resistant materials for ocean applications and should be consulted. Recommended sources are the Naval Ship Research and Development Center, Annapolis, Maryland, and the Naval Civil Engineering Laboratory, Port Hueneme, California. During the past several years, NCEL has conducted studies which include corrosion and fouling characteristics of ferrous, nonferrous, and plastic materials. Of special interest for designers will be the results of in-situ, long-term, deep-ocean corrosion studies. Also, note section 5.4, General Survey Information, for a listing of candidate materials.

Applicable Documents: MIL-STD-33586, "Metals, Definition of Dissimilar." NAVSHIPS 0900-028-2010, "Material Certification Procedures and Criteria Manual for Manned Non-Combatant Submersibles," and RV-S-0110 (NAVY), "Corrosion Control, Deep Submergence Vehicles," and section 5 of reference 14.

The use of protective marine greases, lubricants, or bedding compounds on mating and sealing surfaces and fasteners is encouraged (see section 5.4).

Applicable Documents: RV-S-0147 (NAVY), "Compounds, Retaining, Application of."

1.3.5 Marine Fouling/Abrasives

Fouling by marine organisms is less of a problem on relatively short DSV missions than for extended submergence time. Some materials are inherently self-poisoning and resist fouling, such as copper-nickel alloys. However, care must be exercised when using any dissimilar metal couples in a system, due to possible galvanic corrosion problems. Often the best alternative is to coat exposed surfaces with a specification marine antifouling paint.

Applicable Documents: MIL-P-15931B, "Paint, Antifouling, Vinyl-Red (Formula No. 121/63);" MIL-P-16189B, "Paint, Antifouling, Vinyl, Black (Formula No. 129/63);" MIL-P-19451, "Antifouling, Red (cold plastic), Formula No. 105;" MIL-P-19449, "Antifouling, Black (cold plastic), Formula No. 146/50."

Biological population and suspended abrasive particulates in sea water will, especially over extended periods of time, fill clearances; jam mechanical parts; clog lines, valves, filters, or screens; close orifices; and contribute to the deterioration of materials. Bottom coring operations, anchors, vehicle thrusters, or the vehicle itself may stir up silt, sediment, sand, and marine life from the ocean floor. Even where compensator housing ports have been finely screened to the ambient sea water, the entry of abrasive remnants of various shelled animals has been known to cut and puncture elastomeric compensating materials.

Assuming that some microscopic sea life could be introduced to the compensating system with sea-water intrusion, component materials and hydraulic or compensating fluids should not provide nutrient value or habitability for the growth continuation of marine organisms. Sludge and slime contamination will readily clog, corrode, and deteriorate a system.

Dynamic seals at the machinery-sea interface will be subjected to the abrasive action of suspended contaminants in sea water. Moreover, when the equipment is out of the water, seals will dry out and hard salt and silt deposits may occur. Abrasives may also be introduced into the compensating fluid from dielectric breakdown products of the fluid under electrical arcing and from component and d-c brush-wear debris.

1.3.6 Inclination, Vibration, and Shock

Generally, a compensating system should function satisfactorily when inclined at an attitude up to $\pm 45°$ from the horizontal and for $\pm 60°$ roll.

Operating personnel have commented that a DSV must endure considerable vibration and shock when tied down to the deck of a support ship. Especially for increasing sea states, ship roll, pitch, heave, and propulsion vibration can be quite severe. If a system is air-transportable, this vibration loading must also be considered.

Applicable documents: MIL-STD-167 (SHIPS), "Mechanical Vibrations of Shipboard Equipment" and SAE AIR 1063, "General Environmental Requirements for Deep Submersible Vehicles and Submarines," Society of Automotive Engineers, Inc., New York, N.Y.

A compensating system should be designed to withstand three modes of shock: mechanical, fluid, and thermal. Mechanical shock would be that encountered during DSV handling, transportation, launch/recovery operations at the air-sea interface, or striking underwater topography or structures.

Applicable documents: MIL-S-901, "Mechanical Shock of Shipboard Equipment."

Fluid shock within a system will be manifested as pulsations or surges resulting from increased equipment heating during emergency or overload operation, propulsion reversals, fluid gassing, equipment failure, changes in hydraulic system demands (such as starting or stopping of actuators), and ambient pressure transients experienced by compensator elastomeric materials at the air-sea interface.

Thermal shock is not uncommon when transferring a vehicle from air temperatures to sea temperatures. From arctic to tropical conditions, the environmental range is considerable. For example:

- With a 35° to 40° F sea surface, air temperatures can be 100° to 110° F off the California coast in summer.

- In mid-winter, off the Carolina coast, 0° F air and 60° to 70° F sea surface can be encountered.

Inadequate fluid fill and bleed procedures and/or predive compensator-volume positioning for the range of

environmental temperature can result in compensator overpressurization or depletion of compensating volume at depth.

1.3.7 Dynamic Loading

On board a support or surface-handling ship, DSV compensating systems will experience dynamic loading imposed by ship roll, pitch, and heave in sea states up to and including sea state 5. Fully exposed vehicle equipment (manipulator, thruster motors, etc) and associated compensating devices should be designed to withstand the following:

- Dynamic loading such as would be experienced for vehicle tow up to 10 knots in relatively heavy seas.
- Wave slap of 500 psf.
- Icing load - up to 4.5 psf (when applicable).

1.3.8 Heat Dissipation

Although some enclosure materials are light in weight, nonconducting electrically, chemically inert, and quite compatible environmentally, often their heat-transfer characteristics leave much to be desired. Fiber glass, weldable polyvinyl chloride, and various reinforced and unreinforced plastics will perhaps fall into this category. Whereas these materials may perform somewhat as thermal insulators, they are suitable for electronic and other enclosures where very low-to-moderate heating occurs. Materials such as hard anodized 6061-T6 aluminum and light-gage type 316 stainless steel provide good heat transfer, moderate strength, and low weight.

There is some tendency for compensating fluids of higher viscosity to form a thermal boundary layer at interior enclosure surfaces. Those of lower viscosity, with low internal frictional resistance, higher specific heat, and higher thermal conductivity exhibit better heat transfer properties for low flow rates or static conditions. Also, the increase of fluid viscosity with pressure will tend to increase power losses in rotating machinery, and additional heat will be generated. Since rotating machinery of high horsepower-to-weight ratio is desirable, this imposes a limitation on heat-transfer surfaces and compensating-fluid volume. Therefore, where feasible, fluid-circulation devices are recommended to aid in the distribution and dissipation of heat throughout the equipment housing.

For small, deep-diving submersibles, cruise speed may range from 1 to 3 knots. Although the ocean provides a tremendous heat sink, most DSV machinery systems will be located within hydrodynamic fairings of the vehicle. Thus, equipment enclosures should be designed to provide adequate heat transfer for essentially static external conditions (no cooling water flow). During DSV checkouts, if it is necessary to operate hydraulic or other power systems in air for any length of time, auxiliary cooling equipment will likely be required.

1.4 SYSTEM DESIGN OBJECTIVES AND PHILOSOPHY

1.4.1 Reliability and Safety

Under all operating conditions, the contamination, malfunction, failure, or emergency release of a manipulator or an auxiliary system should not incur damage to critical DSV systems. Opportunity for sea-water intrusion and/or leakage of hydraulic or compensating fluid must be minimized. Systems should require a minimum number of static seals and thorough consideration must be given to the design of dynamic seal systems (see DOT Seals Handbook,[34]). For critical systems of manned submersibles, redundancy or backup systems are also required to assure personnel safety. Thus, system failure mode and effect analyses are very useful. These considerations are of high importance when evaluating equipment "tradeoffs" and the reliability of conceptual system designs.

Systems should be subjected to preoperational hydrostatic testing to ensure that equipment will function in a reliable, safe, and efficient manner without fluid exchange or leakage. For simulated operational testing, equipment should function satisfactorily with operating pressure internal, test depth pressure external (1.5 safety factor on maximum operational), and preferably at lowest anticipated ambient water temperatures (say 35° to 40° F.)

Compatibility with the environment, simplicity of design, and fail-sick or fail-safe features are stressed for both compensating systems and operating machinery. To avoid rupture or malfunction of hydraulic components, systems should be designed to relieve trapped fluid pressure during ascent and descent modes. For emergency deballasting, the interfacing of certain auxiliary equipment on modules should facilitate release or jettison. If massive sea-water intrusion occurs, due to a seal or compensator failure, machinery systems should be capable of emergency operation in a sea-water-flooded

condition for a minimum of one duty cycle before appreciable deterioration in performance appears. These design requirements will further enhance vehicle safety and reliability.

1.4.2 Maintainability and Adequacy of Operating Life

Considering logistic support, a DSV will either be independent or surface-ship-oriented (dependent upon surface ship for support and transportation). An independent submersible enjoys relative freedom from operational restrictions such as weather, range, and duration. Therefore, by definition, a self-sufficient vehicle will be larger, more seaworthy, and all systems must be designed for long, maintenance-free operating life. Long endurance, and thus long maintenance intervals, will place more stringent design requirements on all machinery and compensating systems. To ensure compatibility with the environment over an extended submergence time, a conservative approach is required in areas such as fluid selection, electrical insulation, seals, compensator sizing and protective devices, and general system design.

Small, deep-diving submersibles, whether manned or unmanned, tethered or untethered, are necessarily surface-ship-oriented and operating time for a typical mission is often only a few hours. All vehicle systems will usually receive both a predive and postdive checkout. Therefore, compensating fluids and systems will be inspected and maintained between short-term submersible missions. For instance, a fluid-filled d-c drive motor may have portions of the electrical insulation system which are incompatible with sea-water contamination. One maintenance philosophy is to employ a basically nonemulsifying compensating fluid so that any sea-water intrusion will tend to settle out of the fluid to the enclosure sump where it may be drained off during the postdive checkout or maintenance.

Wherever practical, subassembly system design is encouraged to facilitate removal from the vehicle for maintenance. Modular packaging of hydraulic or other system components may provide space and weight savings and increased overall reliability by permitting direct substitution with spare modular units during maintenance or overhaul. When compatible with design constraints, strive for standardization and duplication of vehicle compensating devices, relief valves, etc, for ease of maintenance and interchangeability of replacement parts.

Fluid-filled enclosures should be designed for maximum accessibility to the components contained within

them. Moreover, the placement and physical arrangement of hardware and compensators should provide accessibility for general inspection and maintenance, fluid sampling, fill/drain operations, or in-place fluid reconditioning by means of an external purification unit. Fluid ports and quick-disconnects should be strategically located to best facilitate all fluid maintenance. Enclosure designs with sloping bottoms or collection sumps are desirable to ensure thorough draining of sludge, sea water, or other fluid contaminants.

1.4.3 Performance and Efficiency

To conserve the limited power available, equipment efficiency is of high priority. Optimum system designs tend to drain minimum amounts of power during periods of minimum work. Input power to equipment should be as low as possible and yet consistent with system reliability and safety. Power transfer must be as efficient as possible.

Although redundant dynamic seals, internal seal pressurization, actuator rod scraper rings and other concepts may be desirable to protect against external fluid contamination, power losses incurred must be minimal. As mentioned previously, a design goal is that machinery should be operable in a sea-water-flooded emergency mode for a minimum of one duty cycle before appreciable deterioration in performance occurs.

For the total range of volume change or travel in compensating devices, fluid "dead-band" on inactive fluid should be minimal. Compensating devices must readily respond to all changes in fluid volume so that the system will not experience negative transient pressure differentials.

1.4.4 Space and Weight

As operational depths increase, space and weight requirements become more critical. System design and physical arrangement must be compatible with overall vehicle arrangement and weight objectives. Component weight influences the buoyancy balance of a vehicle; unscheduled overweight cannot be tolerated. Thus, systems which provide high ratios of power to space and to weight are desirable for this application. HOWEVER, THE RELIABILITY AND OPERATIONAL LIFE OF CRITICAL SYSTEMS MUST NOT BE JEOPARDIZED FOR THE SOLE PURPOSE OF WEIGHT AND SPACE SAVINGS.

Modular and subassembly construction techniques have been shown to provide substantial space and weight savings as well as superior reliability. A modular-hydraulics concept involves the combination of compatible, carefully matched components in a single, lightweight housing incorporating most of the system plumbing internally with drilled and cored fluid passages. Within hydraulic system enclosures, modular valve packages can reduce space requirements by the deletion of individual component foundations and tubing runs. The required intercomponent tubing is decreased and simplified; the number of fluid connections, seals, and tubing hangers or anchors is also reduced. Thus, system reliability may be enhanced. This technique also represents potential space and weight savings due to reductions in enclosure size and compensating fluid volume.

System physical arrangement should be optimized. Whereas careful planning in the relative location of components will reduce both the amount and size of hydraulic tubing, potential advantages are reductions in fluid flow losses, system fluid volume, and compensating volume.

In a broad sense, a modular concept can be thought of as a "building block" construction technique with the objective of containing more of a system's components within a single, compact mechanical package or fluid-filled enclosure. Individual system and subsystem modular units are arranged in the vehicle "space frame" with constraints of overall physical size, weight, electromechanical/hydraulic interfacing, and structural anchors. From the standpoint of system design optimization, backup modules may be continually improved within specified constraints to achieve maximum efficiency and reliability, low power consumption, reduced weight, most efficient utilization of space, etc. The use of modular equipment structures is also encouraged for easier accessibility and to simplify installation, maintenance, and removal procedures.

1.4.5 System Seals/Pressurized Versus Unpressurized Compensation

For fluid-filled deep ocean systems, static and dynamic enclosure seals must prevent both the escape of compensating fluid and the intrusion of the ambient environment, whether in air or submerged. Little data have been available for the use of off-the-shelf dynamic shaft seals in these applications. Also, many design variations of available dynamic seals have made it difficult to select a suitable seal design and seal

system configuration (i.e., single or multiple seals, with or without a pressure differential across the seal), to determine leakage rates for adequately sizing compensating devices, or to establish maintenance schedules. The "Rotary Shaft Seal Selection Handbook for Pressure-Equalized Deep Ocean Equipment"[34] has identified problem areas of existing applications and dynamic seal design variables; guidelines are provided in the selection and application of seals for pressure-compensated rotating machinery.

A good seal system obviously requires a good compensating system (and the converse). Frequently DSV seal problems are directly related to basic seal system-compensating system design deficiencies. Aside from inherent features of certain seals, fundamental consideration must be given to the ramifications of system operating characteristics on compensating devices and all seal configurations.

One design philosophy is to maintain a slight positive bias (Δp) on the compensating fluid to keep the ocean out of the compensating system. Internal pressurization may be achieved by the use of mechanical springs or by the physical location and design of an unbiased elastomeric compensating device so that a static head will be developed due to the difference in specific gravities of the compensating fluid and sea water (see section 2.3.6, Desirable Design Features). Compensator pressurization is not always worth the design effort and complexity. In fact, simplicity of design is considered to be a major contributing factor to system reliability, maintainability, and adequacy of operating life. For spring-biased rolling diaphragm or piston-type compensators, it is difficult to design a compression coil spring which maintains a low, fairly constant loading and good alignment over the range of travel. Sea-water corrosion, marine fouling, abrasive contaminants, and/or spring misalignment aggravate the tendency for mechanical springs to bind. Compensator response can then become erratic, which may lead to overpressurization, negative Δp, or diaphragm failure.

As previously mentioned, at the air-sea interface compensating systems will likely experience both thermal shock and fluid shock, either into or out of sea water. As elastomeric compensator materials and all system seals respond to ambient pressure transients, air or sea-water intrusion into the system may occur. Entrained air in a system will encourage corrosion and the compensating volume will be decreased as increasing ambient sea-water pressure forces fluid into all air voids. This phenomenon is particularly critical for

enclosures which contain rotating shaft penetrations. Therefore, as a desirable design feature for rotating machinery applications, a slight positive internal bias serves to protectively pressurize dynamic seals for atmospheric operation, transportation, and support ship handling as well as for undersea operation.

Electrical distribution boxes, motor controller housings, and enclosures which do not contain moving-shaft seals are not generally pressurized. Internal compensating fluid pressure is equalized with ambient sea pressure by means of a flexible membrane, diaphragm, tube, bladder, or bellows. Where flat gasket-sealing concepts are used, careful attention must be given to the placement, spacing, and torque of gasket fasteners (consult Machine Design Seals Reference Issue and SAE and ASME design standards). At maximum depth, ambient compressive forces on flat elastomeric gasket materials can exceed installation sealing stresses, resulting in unseating of the gasket.

Leakage continues to be a problem for low-pressure differential static seals in a high ambient pressure fluid environment. There is a tendency for an O-ring seal to "walk" from one edge of a rectangular O-ring groove to the other under pressure cycling. This deforming and twisting of the O-ring in the groove appears to pump from one fluid media to the other. Components have also been known to fail because an O-ring in a delta groove has extruded because of trapped fluid introduced during maximum ambient conditions. Under high ambient pressures, O-ring precompression or "footprint" may be considerably reduced due to the bulk modulus change of the O-ring material (see O-ring footprint evaluation[17].) Also, during depressurization, especially at the low end of a pressure cycle, insufficient O-ring "squeeze" and/or bulk modulus hysteresis can cause fluid leakage. Considering many of these characteristics, the use of dual O-ring seals for low-pressure differential enclosure and penetrator applications is somewhat questionable. In this configuration, O-ring sealing ability is not enhanced by the surrounding high pressure; it is critically dependent upon initial O-ring squeeze. Fluid which may become trapped between dual O-rings during pressure cycling will encourage O-ring extrusion and leakage. The use of dual O-rings should also be avoided for reciprocating applications, such as actuator rods, unless a pressure-compensated interseal cavity is employed. Otherwise, fluid "pumping," heating, and the buildup of extreme pressures between the O-rings will impose excessive equipment loads and high O-ring wear rates. Therefore, the combined effects of seal compounds, hardness, squeeze, compatibility, etc deserve

meticulous attention. O-rings and all seal materials should conform to certification specifications for the particular application and should be thoroughly inspected for flaws or voids (by X-ray, if necessary). Ample time should be allocated for preoperational system pressure testing to detect seal deficiencies.

Even though a compensating system may be pressure biased positively, with a higher pressure on the fluid side than the ambient environment, sea water has been known to invade the compensating fluid. The operability and reliability of a system may be reduced considerably. It is difficult to predict the performance of a rotating shaft seal, but hydrodynamic pumping effects have been observed in contact seals which operate under low or even zero pressure differentials. Pumping leakage can also take place against the pressure differential.

The feasibility of pressure-compensated, deep ocean rotating machinery systems depends on the performance of dynamic seals at the rotating shaft penetrations. As mentioned in the DOT Seals Handbook, contact seals (face, lip, and slipper type) may be considered narrow journal bearings which normally have minimal side leakage. Tests conducted by the laboratory at Annapolis have revealed that for 0 to 5 psi internal-to-external pressure differential, properly designed and installed rotary shaft seals at liquid-liquid interfaces do not incur significant leakage or local fluid mixing. Some factors which significantly encourage outward or inward seal "pumping" and leakage are:

- Shaft misalignment, eccentricity, axial and radial runout, vibration, and surface waviness.
- A deteriorated shaft bearing condition.
- Seal elastomer damage from sea water and compensating fluid particle contaminants.
- Underbalancing of the seal due to pressure applied in the reverse direction for which the seal was designed or excessive pressure differential in either direction.
- Combined effects of seal geometry and closing force, fluid viscosity, interface clearance of seal surfaces, elastomer compressibility, etc.

Hydrostatic leakage for low pressure differential contact seals is not significant. Contact seals have an opening force which is due mainly to hydrodynamic effects. Closing force is dependent on seal spring and

elastic forces and should be manipulated to best suit each application. Seal breakout torque (at start-up) should be low to have minimal static and dynamic leakage.

Tests at Annapolis also indicate that lip seals may be undesirable for applications at the machinery-sea interface, due to inherent features of the lip seal design. While in continual operation, "mixing leakage" is minimal; however, when the shaft comes to rest, there is a tendency for abrasive particles and sea water to penetrate the seal clearance area as the seal cools and contracts. At system start-up, sealing ability rapidly deteriorates due to the action of abrasives which are trapped between the shaft and the lip of the seal.

Shaft-sealing concepts at the machinery/sea interface should be considered with the following factors in mind:

- A reasonably practical rotating shaft seal arrangement will <u>not</u> provide zero leakage.

- For drive system machinery design and materials selection, consider at least minimal sea-water contamination (up to 2%).

- Compensator design and protective features should preclude excessive internal pressure from distorting or "blowing out" the shaft seal.

- Assuming that a given contact seal has been correctly designed and installed, some slight positive bias does not seem to enhance the sealing ability nor does it increase the tendency for a "good" seal to leak compensating fluid.

- The susceptibility of a dynamic shaft seal to sea-water and grit intrusion appears to be directly related to the amount of <u>seal clearance</u>. Although in conflict with conventional seal design practice, for this application it may be necessary to design a dynamic seal which has no ability to generate a clearance.

- For an unpressurized compensating system, there is the possibility of a small pressure differential in either direction; thus, a bidirectional shaft seal should be employed. Reversed pressures can result in catastrophic failure of a unidirectional contact seal.

When the system design dictates a positive fluid pressure on dynamic shaft seals, the following rationale is applicable:

- Minimum differential pressure will allow the longest contamination-free operating time following a condition which initiated significant leakage.

- A positive Δp differential should be the minimum possible and yet offset transient pressure reversals which occur during DSV transportation or launch/recovery operations. An inward pressure reversal can provide an opportunity for ambient air or sea water to penetrate dynamic seals.

- Considering the various hostile aspects of the sea-water environment, a pressurized compensating device should be mechanically simple and as "fool-proof" as possible. Piston devices or mechanical springs should not tend to misalign or bind. The reliability of a compensating system, however, should never be jeopardized for the sake of pressurization.

- A positive Δp up to 5 psi should be adequate to compensate a system for both atmospheric and depth pressure while seal breakout torque and power losses will remain low. To withstand increased internal pressurization above 5 psi, enclosures for the equipment would likely incur a considerable weight penalty.

The question arises whether to use one or two dynamic seals at rotating shaft penetrations. For smaller, less complex DSV's and work systems, single dynamic seal designs are often used. The rationale is that weight is a critical factor and that machinery will be inspected and maintained on a short-term basis, between dives. A different set of ground rules must be imposed for long-term submergence. Systems are necessarily more conservatively designed and long operational life and reliability take priority over the weight penalty or system complexity. At the machinery-sea interface, a dual-seal arrangement is employed. The outboard and inboard shaft seals are separated by a fluid-filled, pressure-compensated interseal cavity. For either short- or long-term submergence, a dual-seal arrangement is recommended for speed reducers such as would be used with fluid-filled electric drive motors (see section 2.6).

Linear hydraulic actuators which were developed for DSRV-I have employed a patented variation of this protective, dual-seal concept with a high degree of

success. For a description of this actuator piston-rod seal design, see article on page 84, "Hydraulics & Pneumatics," December 1970. Also, refer to chapter II for a discussion of compensating schemes and seal systems for rotating machinery.

1.5 COMPENSATING SYSTEM INADEQUACIES

Designers of fluid-filled, pressure-compensated systems have not fully appreciated all design parameters or the ramifications of the physical arrangement of compensating systems. Compensating devices have historically been underdesigned and are often deficient both in fluid volume allowances and protective features. Although there has been a lack of availability of compensating fluids which have all of the properties of an "ideal" compensating fluid, many properties have not been adequately defined in relation to this application.

Trapped air in systems has been a dominant problem. If fluid fill and bleed procedures cannot be improved sufficiently to solve this problem, compensating volume must be increased considerably. Another typical, very basic problem is that potential compensator hardware, as supplied by the manufacturer, is poorly compatible with a hostile, high-pressure ocean environment. Corrosion, abrasion, and marine fouling problems have caused inferior compensator response, fluid leakage, binding, and failure. Inadequacies of various static and dynamic seal designs appear to be a more significant problem than overall compatibility of elastomeric materials.

Too often designers will compromise adequate compensating volume to use a conveniently available elastomeric bellows, diaphragm, or bladder. Although economy of space and weight are important, being frugal with the amount of volume compensation is the least desirable goal.

A lack of fundamental consideration for system operating characteristics has resulted in compensating system designs which reduce system reliability and operational life. A widespread philosophy is to positively pressure bias the compensating fluid so that any leakage would be outward, rather than the inward invasion of sea water. However, analyses of operational failures indicate that most sea-water leakage problems are directly related to seal system-compensating system design deficiencies. Transient fluid shock in hydraulic systems has been intensified by improper circuit design wherein circuit elements may contain uncompensated

volumes in certain modes of operation. Fluid gassing has been found unexpectedly in a-c drive motors. Internal overpressurization from unrelieved gassing has resulted in catastrophic failure of seals, compensators, and equipment.

Aside from relief valving, redundancy in dynamic seals, or cascading the compensation, other concepts to enhance environmental compatibility and reliability are needed in seal system-compensating system designs. If significant outward leakage occurs so that the compensating device becomes depleted, features for admitting sea water as the compensating fluid may be desirable to achieve a "fail-sick" emergency operating condition. Otherwise, system enclosures, seals, or compensating devices are likely to incur catastrophic implosion.

CHAPTER II

COMPENSATING DEVICES AND SYSTEM ARRANGEMENT

2.1 INTRODUCTION

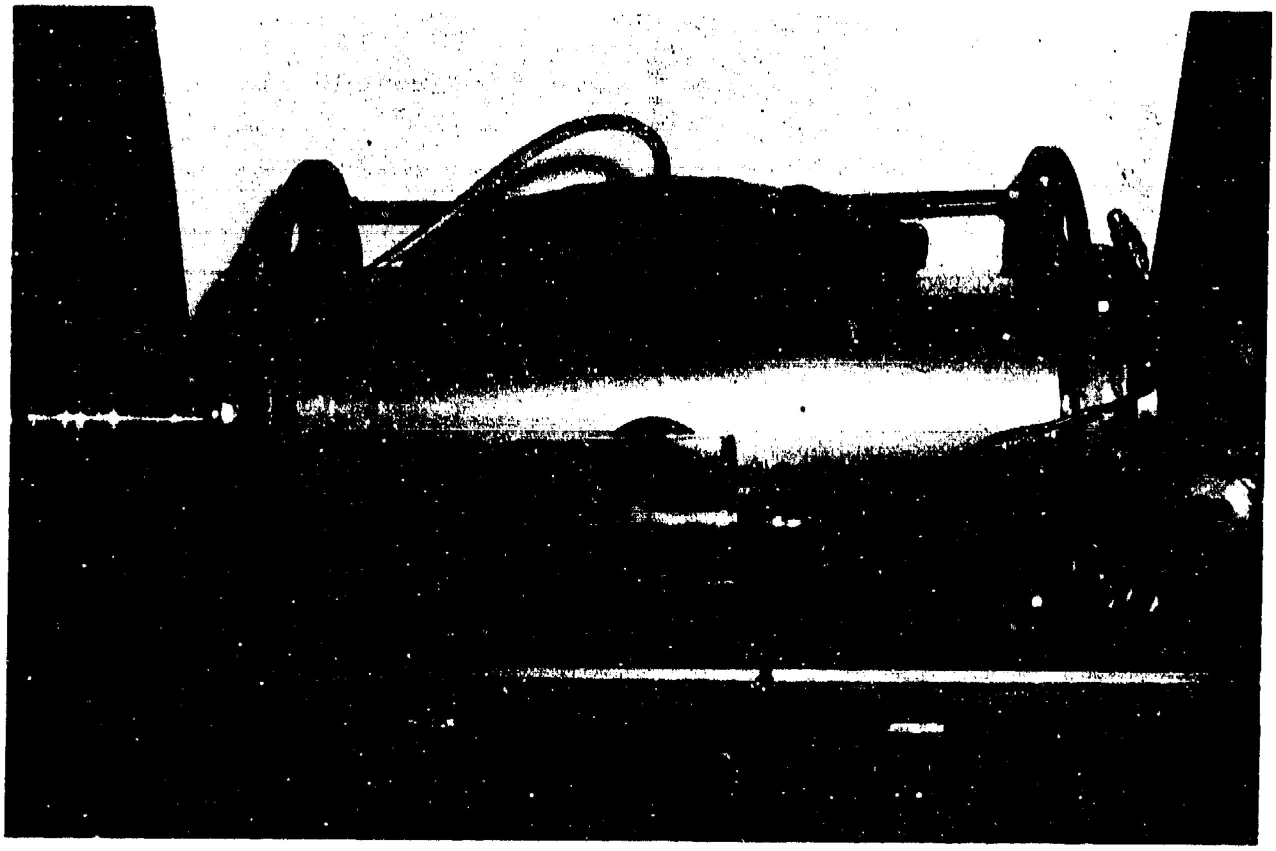
This chapter enumerates many features and configurations of compensating devices and systems which deserve thorough design consideration. While this will serve as groundwork, designers are encouraged to make further deductions and develop innovative compensating schemes. For reliability, a minimum number of compensating devices per system is recommended. There must be a full integration of compensating system design with that of fluid-filled equipments in order to provide vehicle systems which are both very reliable yet practical to maintain.

2.2 COMPENSATING DEVICE CONFIGURATIONS

2.2.1 <u>Flexible Bag or Bladder</u>

This is fabricated of elastomeric material, with or without fabric reinforcement. A typical device is an accumulator bladder with a modified gas valve assembly. It is usually mounted external to equipment enclosures. Purely elastomeric compensators are <u>not</u> self-supporting (see section 2.3.6, page II-18).

Bladder, Mounted in Plastic Housing

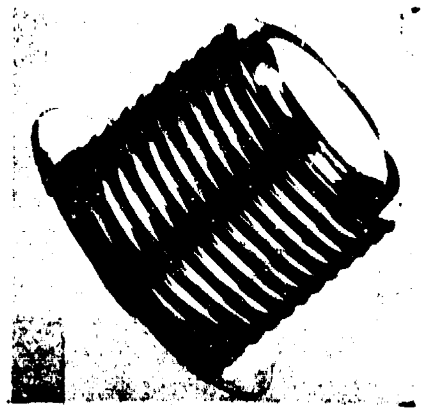


Flexible Bag (Fabric-Reinforced Rubber)

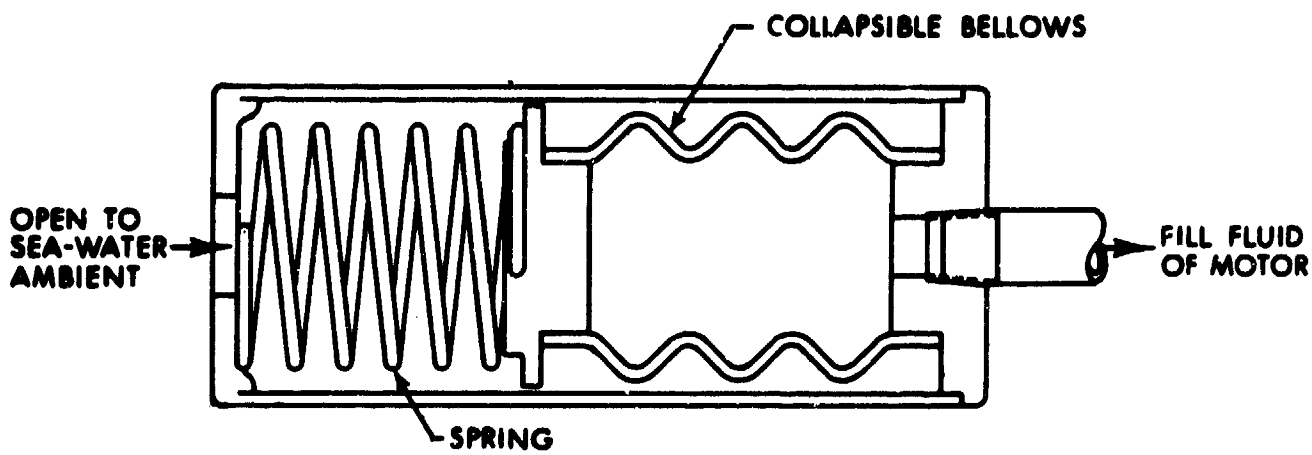
2.2.2 Bellows or Convoluted Tube

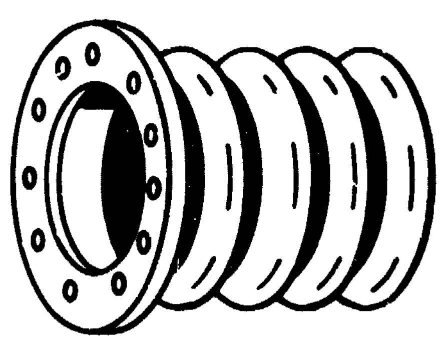
This is fabricated of various metals or elastomers and is suitable for internal or external mounting. Metal bellows are seldom used, perhaps due to expense, limited stroke or volume displacement, and somewhat delicate design for the ocean environment.

Elastomeric bellows and convoluted tubes are fabricated by molding or dipping methods. Positive elasticity of rubber bellows will provide some slight pressurization. If spring loading or other overpressurization is desirable, elastomeric convolutions tend to deform excessively; thus, special design consideration is required. With one application, in order to maintain a rubber bellows configuration under spring loading, PVC rings were placed internal to the bellows and stainless wire hoops reinforced the convolutions externally. For manipulator arms or other applications involving intricate linkages and mechanisms, self-compensating bellows "sleeves" have been employed. This configuration permits mechanical motion and flexure while while all components remain fully immersed in compensating fluid.



Welded Metal Bellows

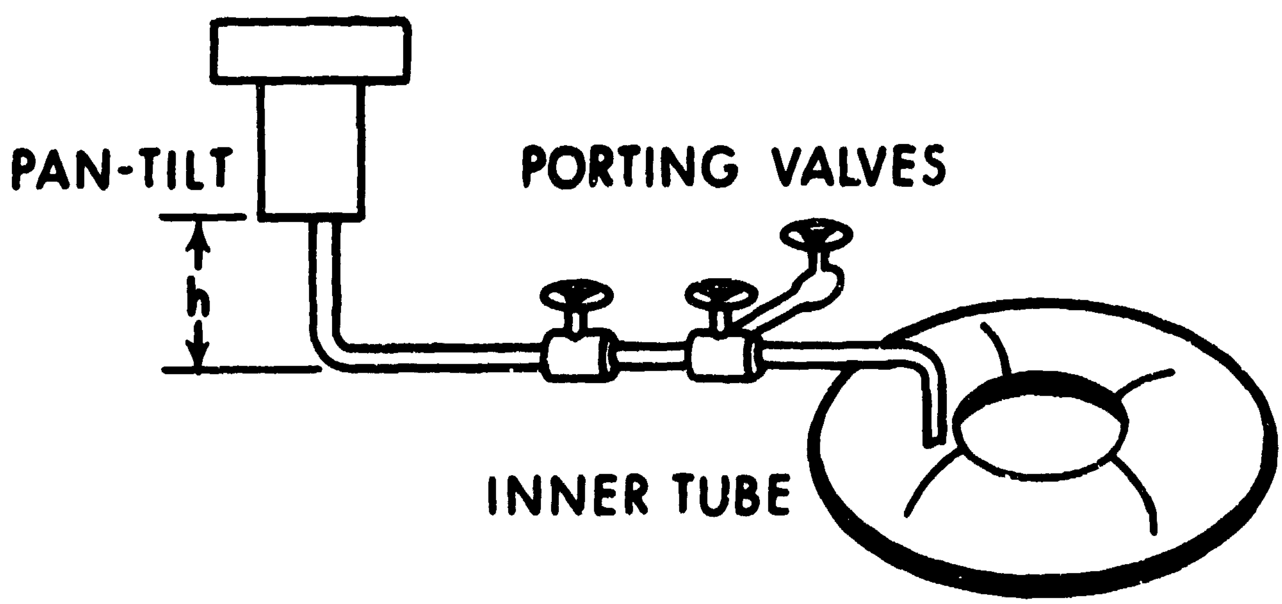


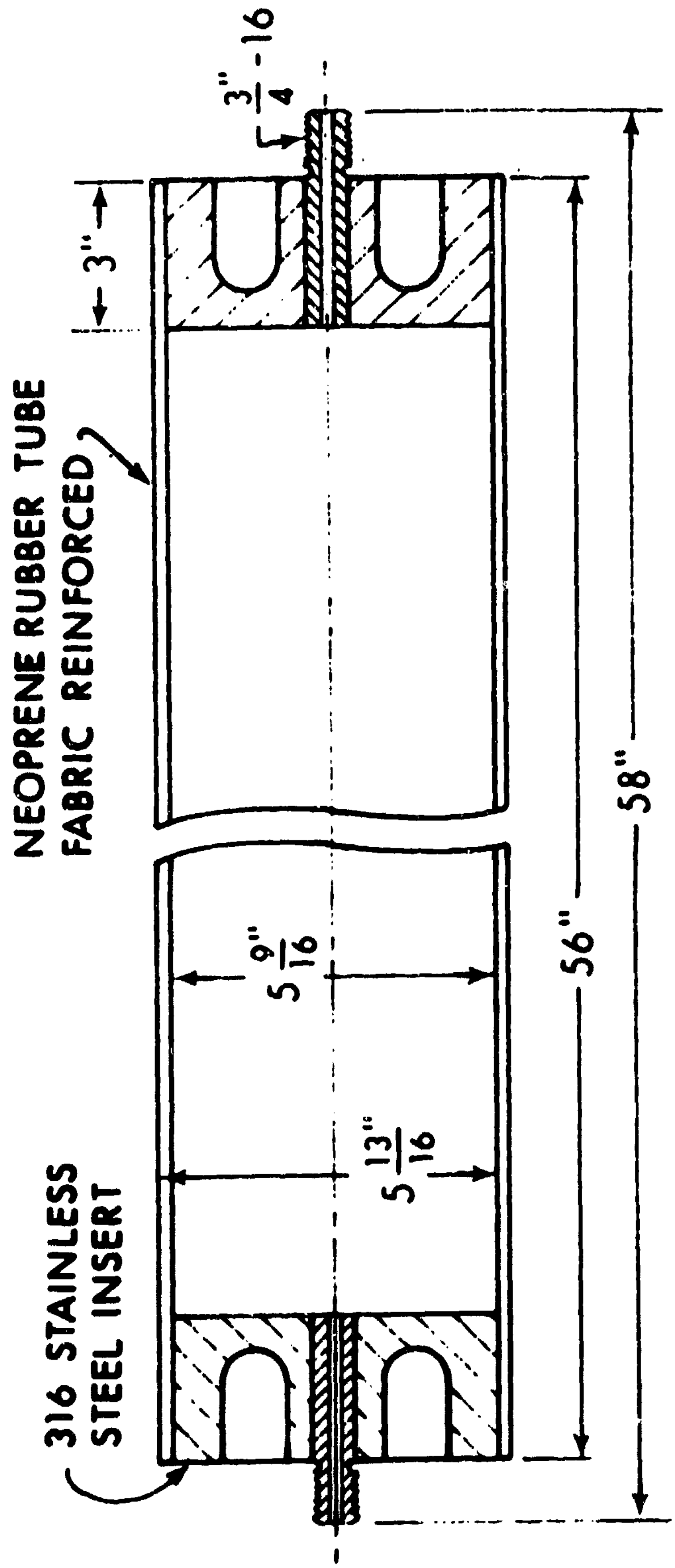


Elastomeric Bellows with Flanged Mounting

2.2.3 Elastomeric Tube

Applications have employed vertical cylindrical tubes with end plugs, circumferential tubes, and even aircraft or motorcycle inner tubes. In a sea-water-flooded a-c motor design, the stator was fluid-filled and compensated by an internally mounted circumferential tube while the rotor was "canned." The pressure of the stator compensating fluid was equalized with that of the ambient by admitting seawater to the inside of the tube. However, since the condition of the tube cannot be periodically assessed without considerable disassembly of the motor, this approach is not recommended.




NEOPRENE RUBBER TUBE
FABRIC REINFORCED
316 STAINLESS
STEEL INSERT
3"
3"/4 -16
58"
56"
5 9/16"
5 13/16"

2.2.4 Diaphragms

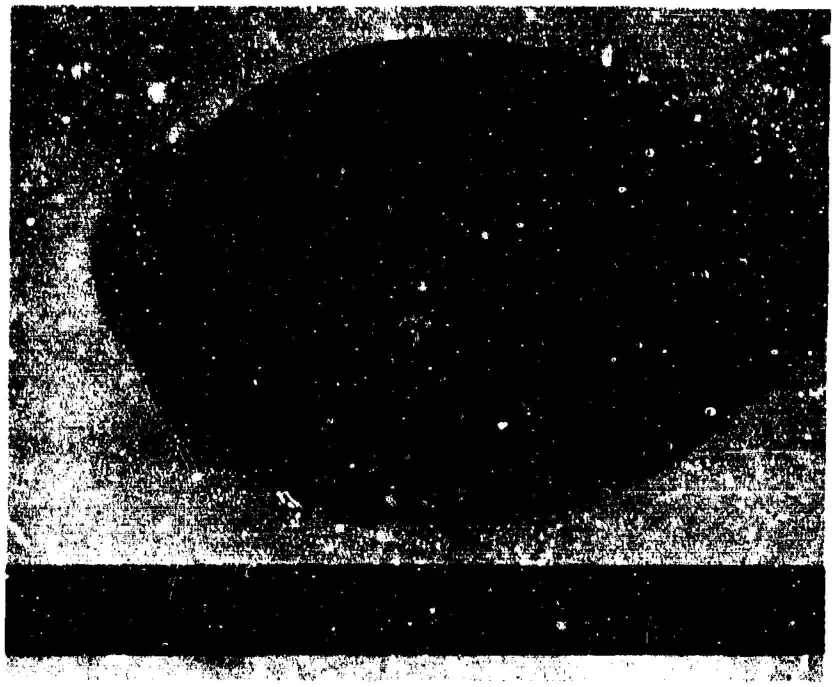
There are three types - flat, convoluted, or rolling diaphragms. Convoluted and rolling diaphragms can be thought of as partially inverted "top hat" diaphragms. Whereas flat and convoluted diaphragms may be either solid or fabric-reinforced elastomeric material, rolling diaphragms are composed of elastomeric-coated, woven fabric material.

The deflection or stroke of a flat diaphragm is directly related to the elastomer stretch over the exposed surface area. For a slight change in fluid volume, flat diaphragms of <u>small</u> surface area can develop considerable positive or negative fluid pressure differentials. A convoluted diaphragm of similar size will provide increased stroke without the constraint of elastomer stretch.

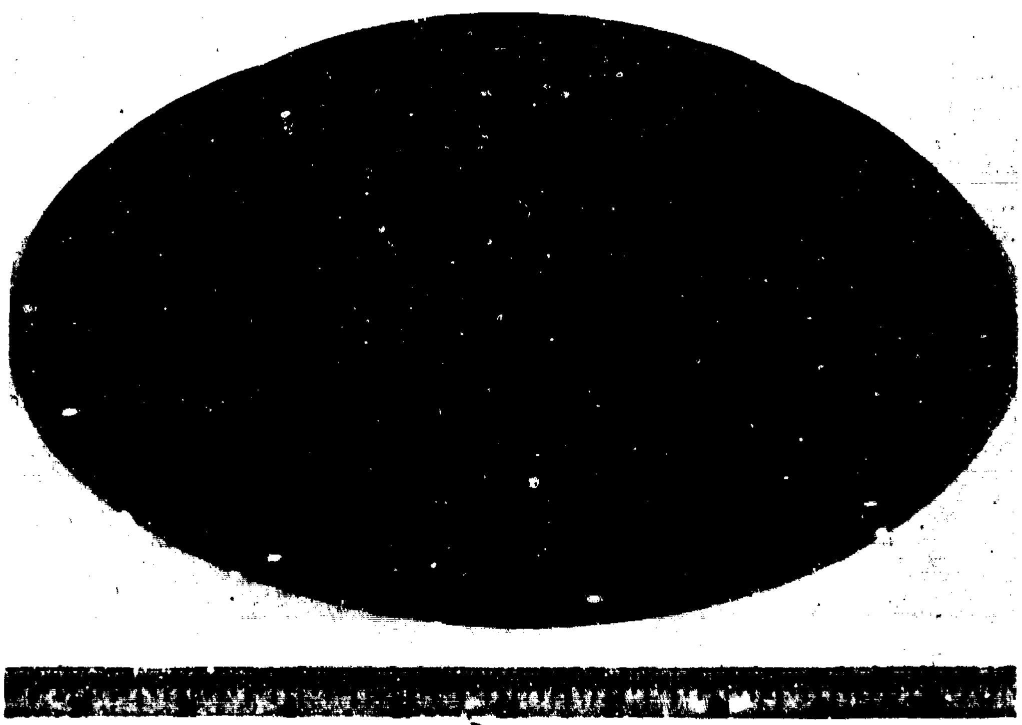
Spring-loaded rolling diaphragms are commonly used in separate, external compensators. Pod-mounted electric drive motor designs have employed integral rolling diaphragm compensating devices. At the antidrive end of the motor, the cylindrical case is extended to house the spring-loaded rolling diaphragm and piston assembly. The thinness and limited abrasion resistance of "standard" rolling diaphragms are considered a disadvantage, especially for applications which involve extended submergence time.

FLAT DIAPHRAGM

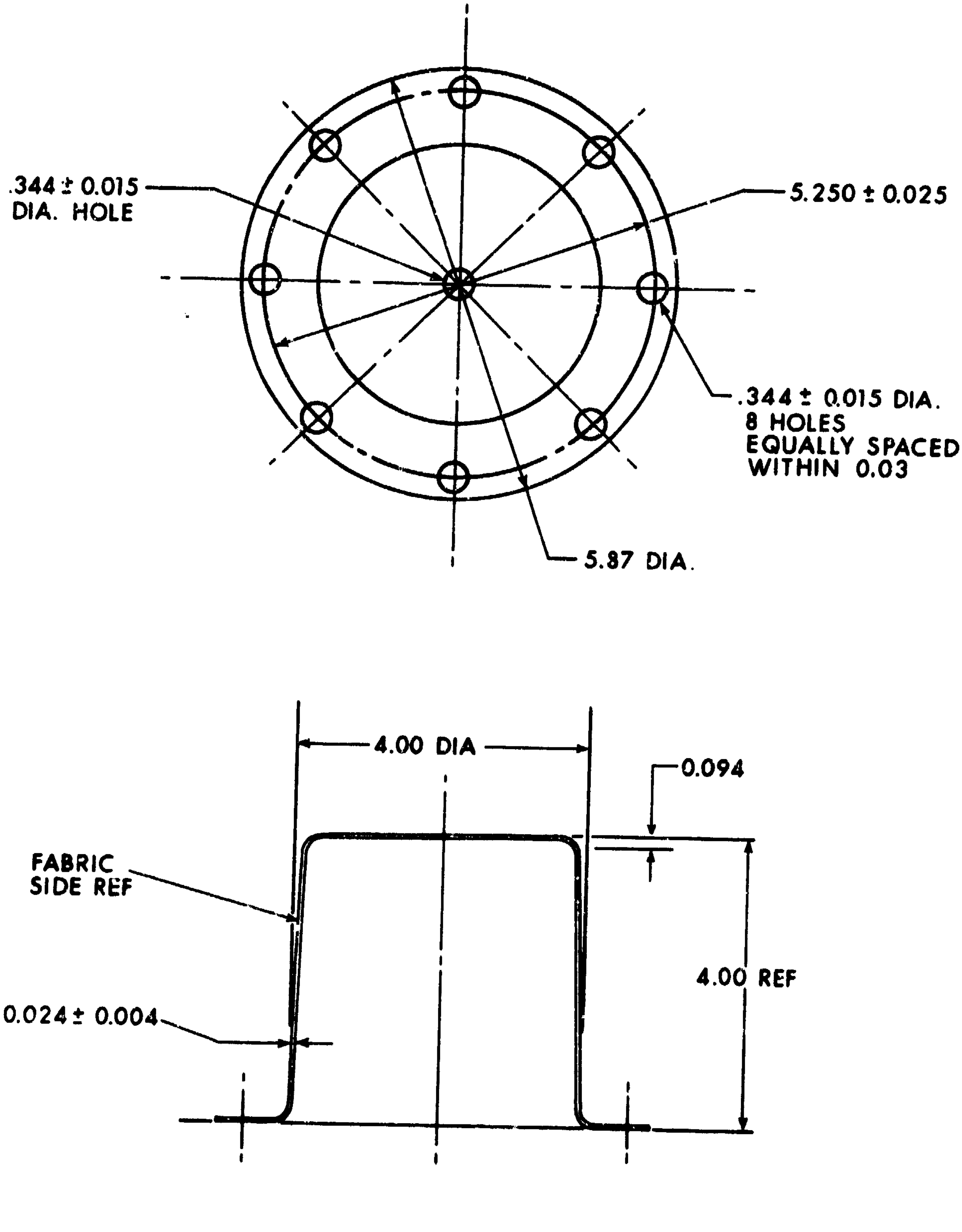
CONVOLUTED DIAPHRAGM



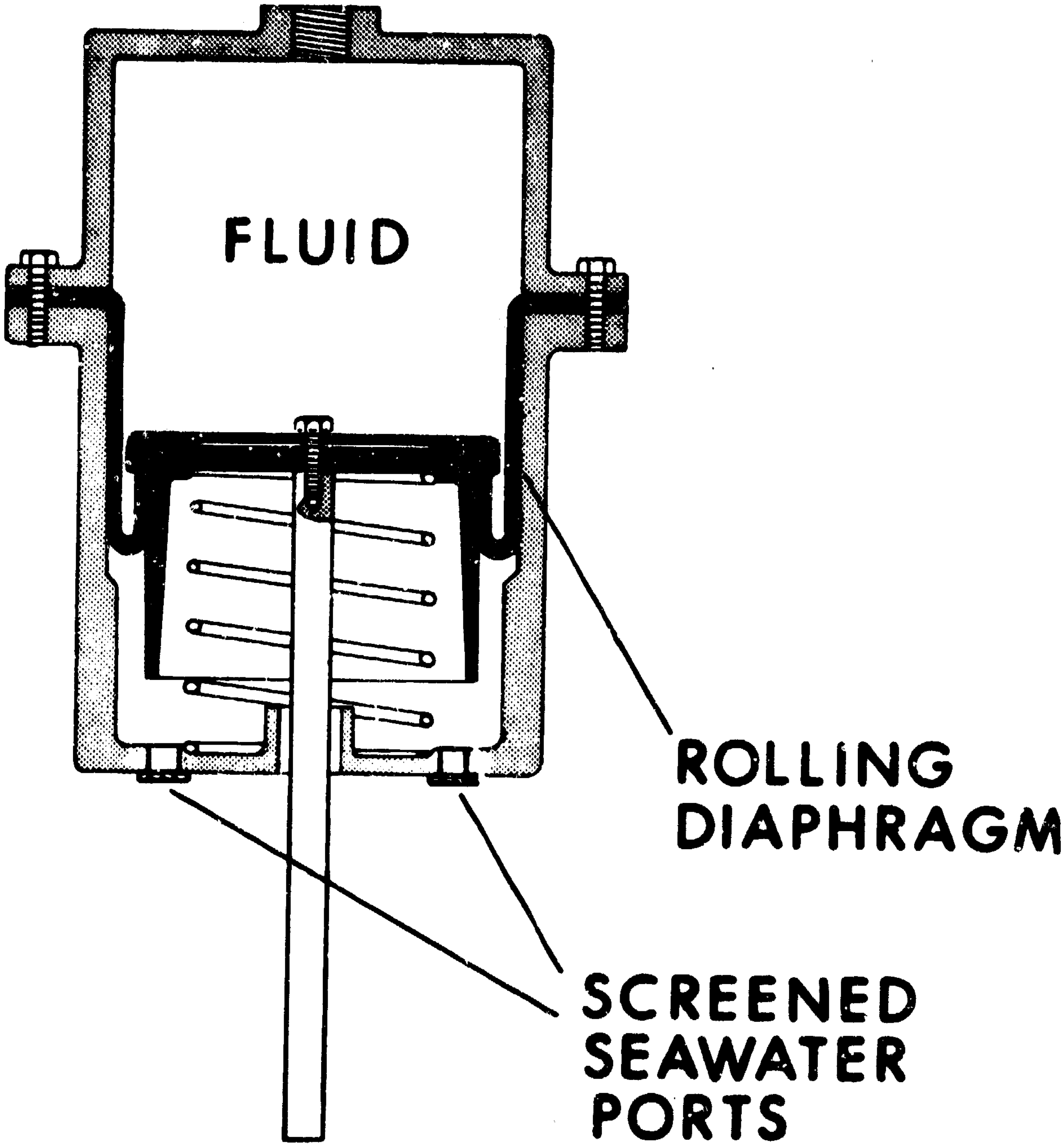
Partially Inverted Top Hat Diaphragm



Convoluted Diaphragm



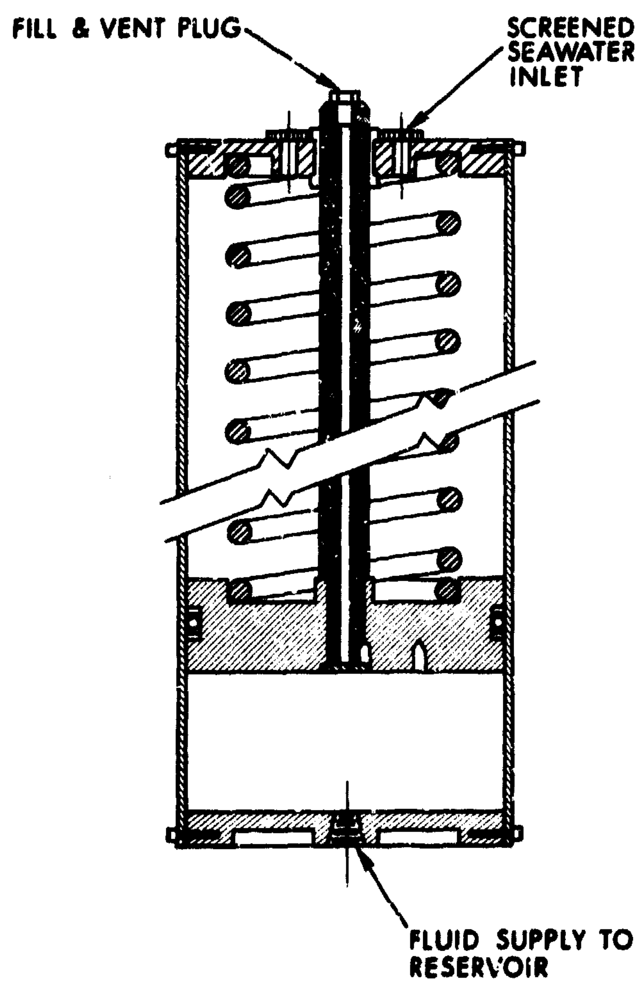
Top-Hat or Fully Extended Rolling Diaphragm



Rolling Diaphragm Compensator

2.2.5 Spring-Loaded Piston

This device is typically a modification of a hydraulic piston accumulator; suitable for external mounting. A cascaded compensating approach is recommended for this configuration so that the outboard side of the piston, dynamic seals, cylinder bore, and spring will not be exposed to sea-water and particle contamination. Dynamic seals which are directly exposed to the ocean environment should be avoided in compensating devices. Not only is "mixing leakage" a potential problem, but contaminants will tend to abrade the piston, cylinder bore, and dynamic seals. Binding of the piston can cause a catastrophic seal-system/compensating-system failure. For example, to cascade the compensation and provide a protective self-adjusting fluid environment, a fluid-filled bladder may be attached to the spring-loaded side of the piston compensator housing.



Spring-Loaded Piston Compensator

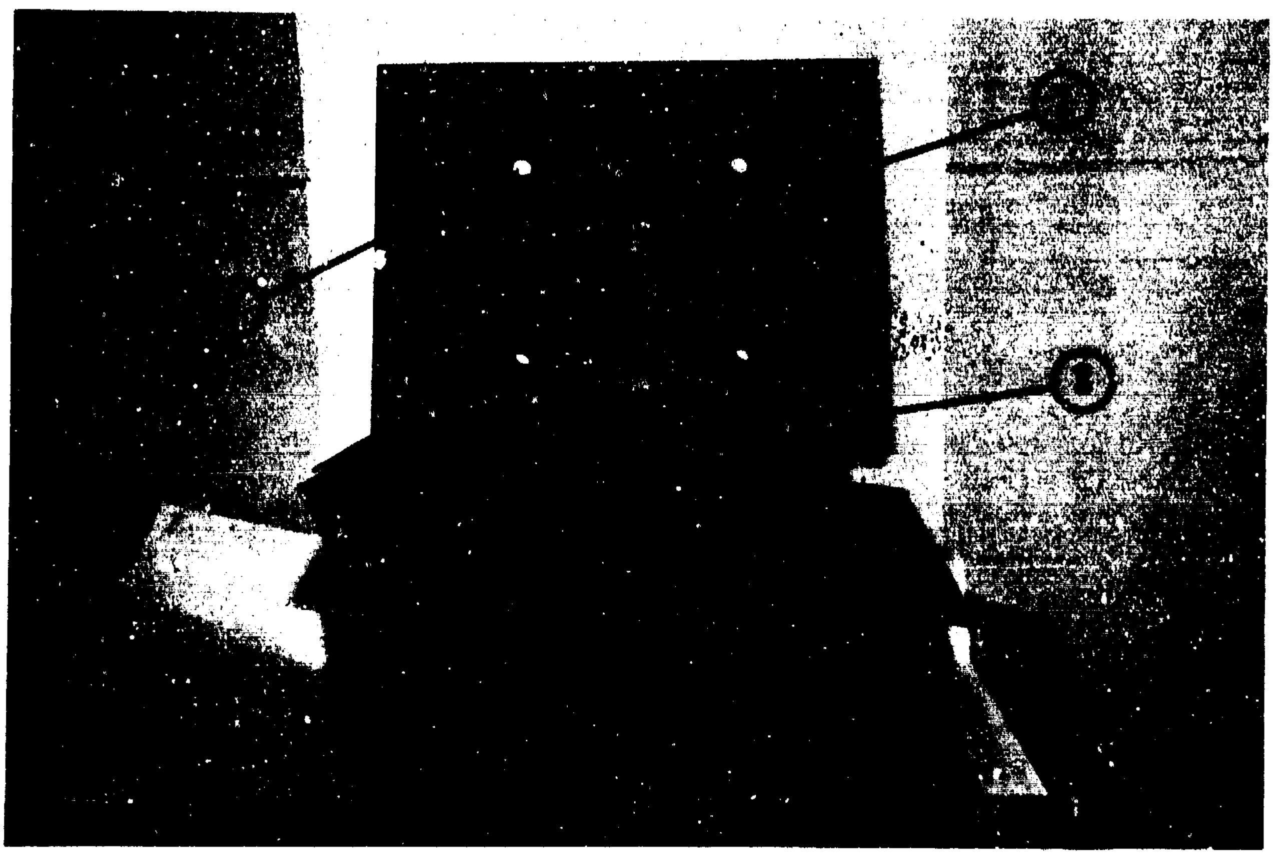
2.2.6 Flexible Membrane

In a broad sense, this concept is an application of a flat elastomeric diaphragm to facilitate a moving interface for one entire wall of an enclosure. This type of enclosure is fabricated with a flanged opening and houses electrical distribution systems or even combined equipment, such as a hydraulic pump-drive system. A ported cover plate or grill is desirable to provide protection, support, and to constrain the membrane from "blowing out" due to overpressurization.

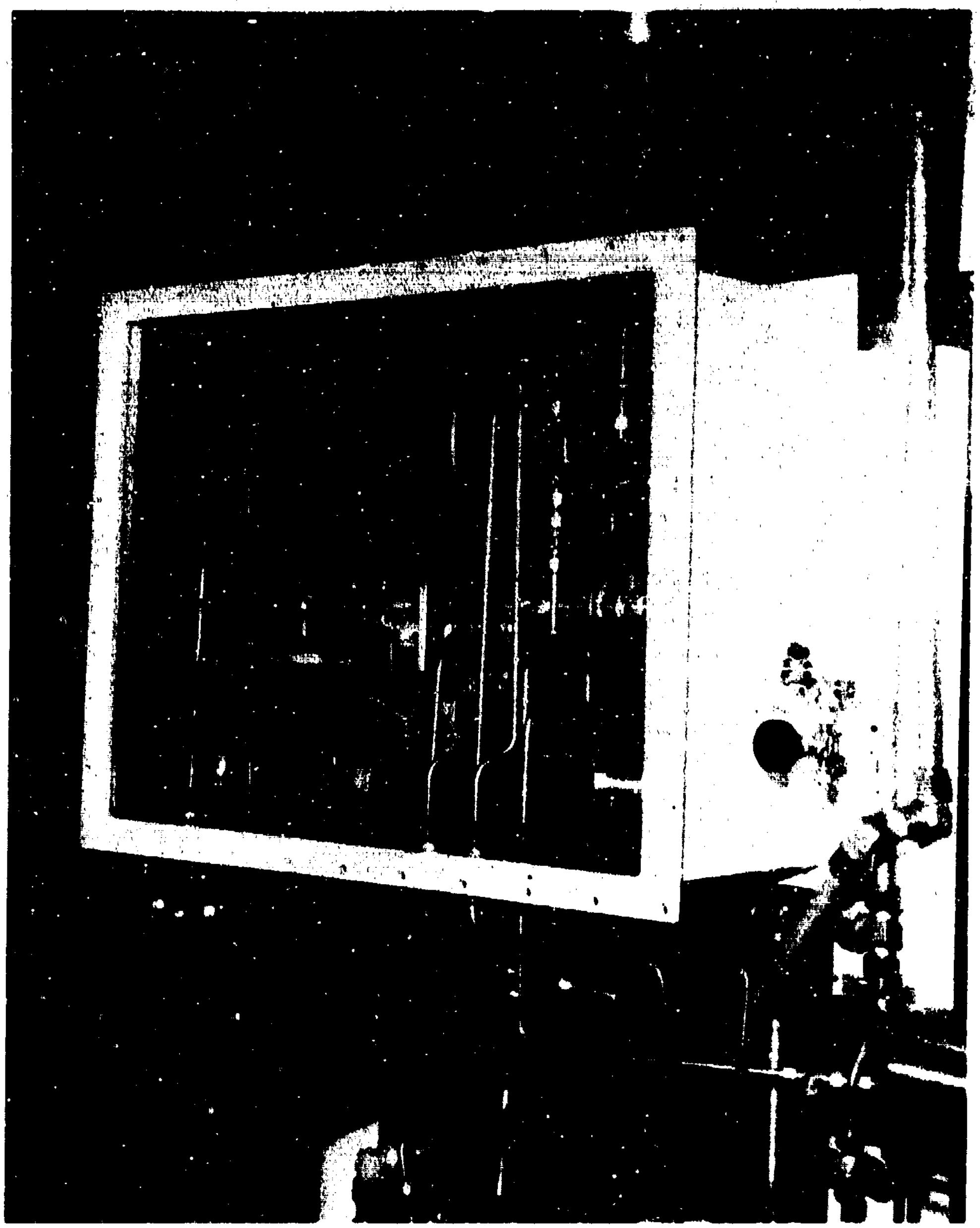
1 - Limber Holes or Sea-Water Ports

2 - Cover Plate

3 - Elastomeric Membrane



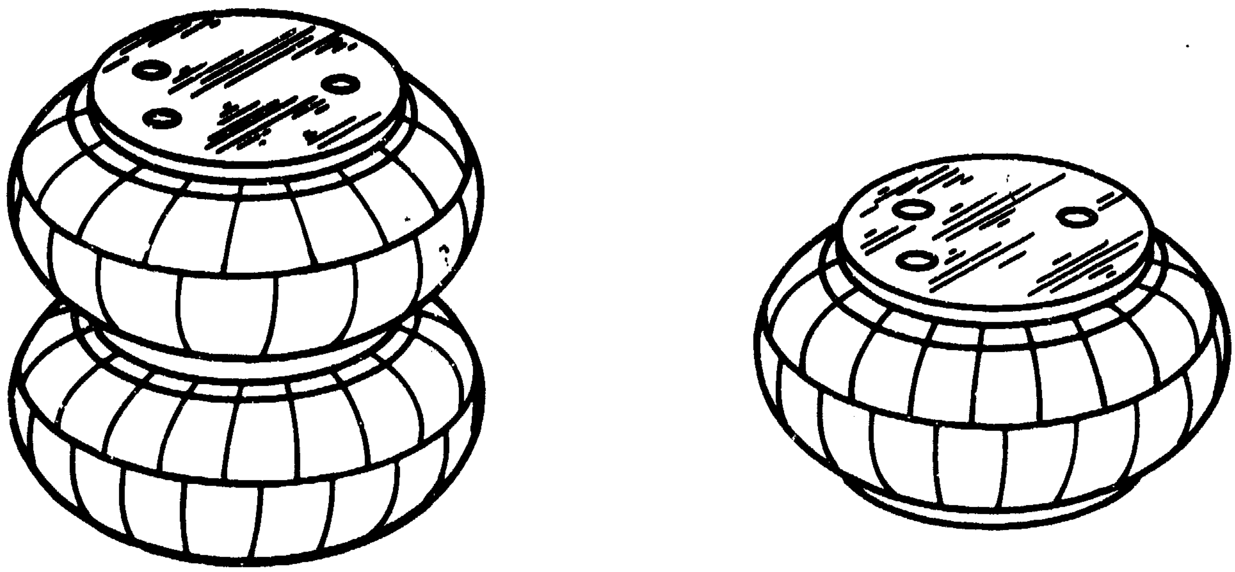
Membrane-Compensated
Hydraulic System Enclosure



Membrane-Compensated Hydraulic System Enclosure

2.2.7 Modified Pneumatic Actuator

This is a rugged variation of a bladder or bellows device which can withstand and maintain considerable internal pressurization and is a possible compensator candidate. Single and multiple convolution designs have been developed. Pneumatic actuators have been filled with air or liquids to either transmit force or to cushion loads and isolate vibration and shock in industrial applications. For some actuators, the use of petroleum-based fluids has not been recommended, apparently due to compatibility problems with fabric-reinforced neoprene materials. Continuing R&D efforts of manufacturers should provide actuators which may be compatible with a wide variety of fluids.

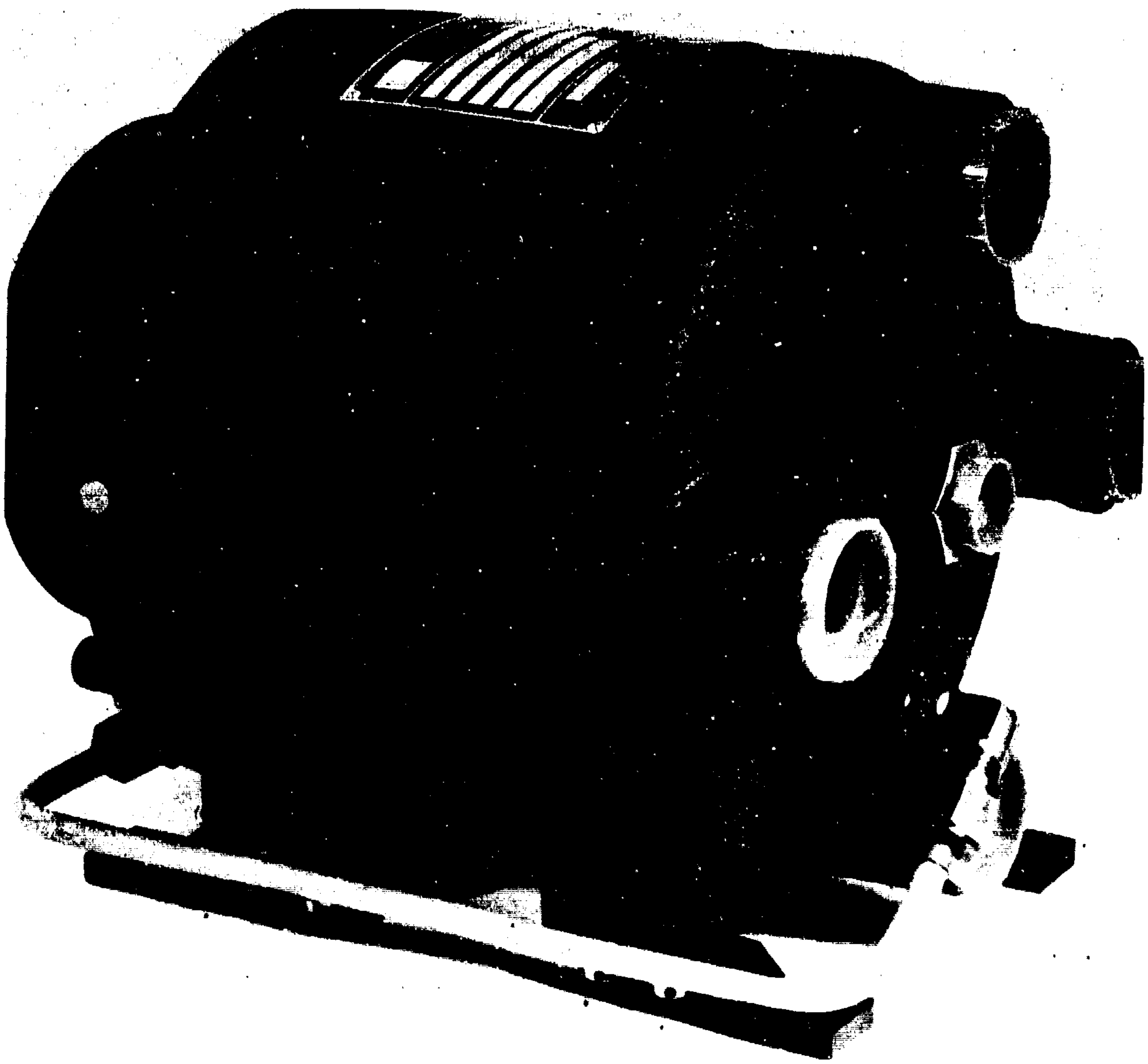


PNEUMATIC ACTUATORS

2.2.8 Self-Pressurizing Reservoir

This is also referred to as a "bootstrap"-type reservoir and has been used extensively in aircraft hydraulic systems. It is readily adaptable for installation in fluid power systems which employ a combined equipment enclosure (section 2.4.2). The surrounding enclosure fluid provides environmental protection and compensates the self-pressurizing reservoir. The use of this device will facilitate a closed hydraulic system arrangement (see section 2.7.3) which is considered very desirable.

Refer to the photograph below and the closed hydraulic system schematic in section 2.7.3, page II-42. Note the central high-pressure port and the four outer low-pressure ports.



2.3 DESIRABLE DESIGN FEATURES FOR COMPENSATING SYSTEMS

2.3.1 Sea-Water Leakage Detectors

Operating personnel may be alerted to the leakage of sea water into vital fluid-filled enclosures and components through the use of a sea-water leak detector system. Detectors are installed in bottom or drain sump areas of enclosures and will short electrically when sea water accumulates and bridges a small electrical gap. In metal enclosures or housings, a detector may consist of a single live electrode or plate. The compensating fluid normally insulates the electrode; an accumulation of sea water will complete the circuit to the metal enclosure bottom (ground). In fiber glass or other electrically nonconductive enclosures, two plates or electrodes, one live and one grounded and closely spaced, may compose a detector (see section 5.4, General Survey Information, for Description of Various Detectors).

A warning light on the vehicle control panel will energize when sea water electrically grounds any leak detector in the system. A selective ohmmeter readout will identify the detector location and verify the warning light indication. Ohmmeter circuitry should be periodically calibrated.

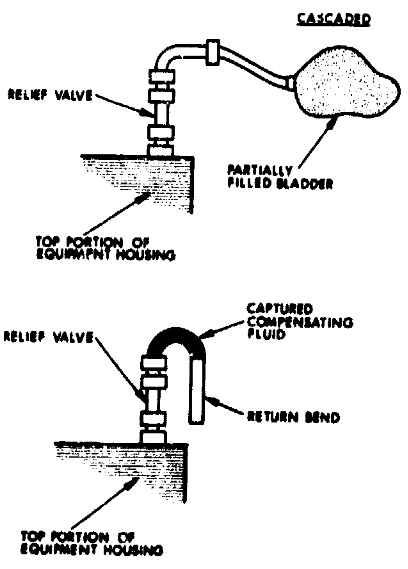
Contaminants other than sea water will also migrate to low points of enclosures. Metal, carbon, or other wear particles and sludge are typical contaminants which may produce unwanted grounding or insulate detector probes. Therefore, according to preventative maintenance schedules, and at all other times when an enclosure is drained and opened for servicing, the insulating and conducting detector surfaces and adjacent enclosure walls should be thoroughly cleaned.

2.3.2 Gassing and Fluid Expansion Relief

To relieve gas generated by the thermal decomposition of a compensating fluid, a relief valve should be located at the highest portion of the equipment enclosure, housing, or compensating system piping. Heat generated by the mechanical action of gears and bearings or electrical hot spots and d-c commutation can reduce a fluid to lower (more volatile) fractions. Both volatile fluids and gases may evolve with the result that electrical insulation and component materials are attacked. If portions of fluid become thermally unstable, especially under conditions of electrical shorting and arcing, gaseous products will also be generated. Unrelieved gassing will displace fluid from higher points of the enclosure or compensating system. Such displacement has been directly responsible for catastrophic equipment failures.

If gassing is not a characteristic of the system, a relief valve should likewise be employed to protect against excessive fluid thermal expansion and internal over-pressurization. A single relief valve will serve to vent either gas or fluid. To determine a valve relief pressure, consider the strength of the compensator elastomeric material, enclosure structure, and maximum rated internal pressure for rotary shaft seals. The relief pressure setting should be below the pressure at which elastomeric materials or seals would leak or rupture. Accordingly, hardware weight will tend to increase with internal pressure rating. Since relief valves normally have some hysteresis, a relief pressure setting which is too low may encourage fluid leakage during pressure transients and/or sea-water aspiration. Low-pressure relief valves are sensitive devices and regular checks should be made to ensure operability and relief pressure setting.

Two methods are shown below for protecting the outboard, discharge port of relief valves from sea-water intrusion. A return bend of tubing will tend to retain a portion of compensating fluid in the tubing bend which provides a barrier against the sea-water environment. Actuation of the relief valve during the fluid fill procedure will insure that fluid is always present in the tubing return bend. Another option is to attach a bladder, partially filled with fluid, to the valve discharge port. This cascaded arrangement allows relieved gas or fluid to collect in the bladder and no portion of the compensating fluid is directly in contact with sea water. If excessive gassing is anticipated, as in the application of "on-off" mechanical contactors, direct relieving to ambient rather than attaching a bladder is recommended.



Relief Valve Protective Features

2.3.3 Protection of Exposed Parts, Surfaces

- Selection of materials, system design, and fabrication should provide low weight, adequate strength, good corrosion resistance, and ease of maintenance. Avoid the use of dissimilar materials which would form galvanic corrosion cells in sea water.

- Consider metal passivation, paints, coatings, protective greases, lubricants, and bedding compounds to minimize corrosion, marine fouling, and enhance the operation and disassembly of the compensating system.

- Use baffles, screens, and other filtering devices to minimize damage from sea-water particle contamination and fouling on elastomer or piston and internal compensator housing. Filters or screens should have large exposed surface areas and moderately sized openings so as not to easily foul and clog.

- Use centrifugal separators, filters, felt pads impregnated with lubricant, or other devices to prevent abrasive particles from damaging outboard rotating shaft seals.

2.3.4 Materials Compatibility

Various metals, alloys, plastics, coatings, and elastomers in the system must be compatible with sea water and/or the compensating fluid. It is also important that the bulk moduli of adjoining or close-fitting materials are approximately the same. The combined effects of temperature, pressure, contaminants, and the softening, shrinkage, "swelling", or compressibility of elastomers will encourage seal deterioration and leakage. Short-term fatigue cracking and checkering of nitrile elastomeric compensator materials has been attributed to ozone attack from air pollutants. General survey information (section 5.4) indicates that nitrile is the most widely used elastomeric material. However, the type of plasticizer used or other compounding variations by a given manufacturer can radically affect the compatibility of an elastomer with a fluid; hence, specific compatibility tests should be conducted. Compatibility data for electrical insulation materials may be found in reference 9. Also refer to section 1.3.4 for additional comments.

2.3.5 Fluid Volume Indicators or Devices to Monitor Compensator Volume

For systems which employ dynamic seals, excessive fluid leakage may occur and some means must be provided to warn operating personnel of low compensator volume.

A guide rod which is connected to the spring-biased piston of a compensator will project out of the piston housing so that the compensator volume may be visually determined. When exposed to the ocean environment, marine fouling and binding of the indicator-guide rod are potential problems. To remotely indicate available compensator volume, fluid-immersed limit switches or magnetic switches are feasible. An LVDT may be used to continuously monitor compensator travel/volume.

Another means to determine that the compensating fluid volume has not been depleted is to monitor differential pressure. Through the use of a strain-gaged metal diaphragm, a differential pressure transducer may be devised to remotely monitor Δp between the compensating fluid and sea water. The strain-gaged diaphragm can be designed to fail electrically at a predetermined maximum Δp (say, 15 psid). Thus the condition of the compensating system can be continually observed.

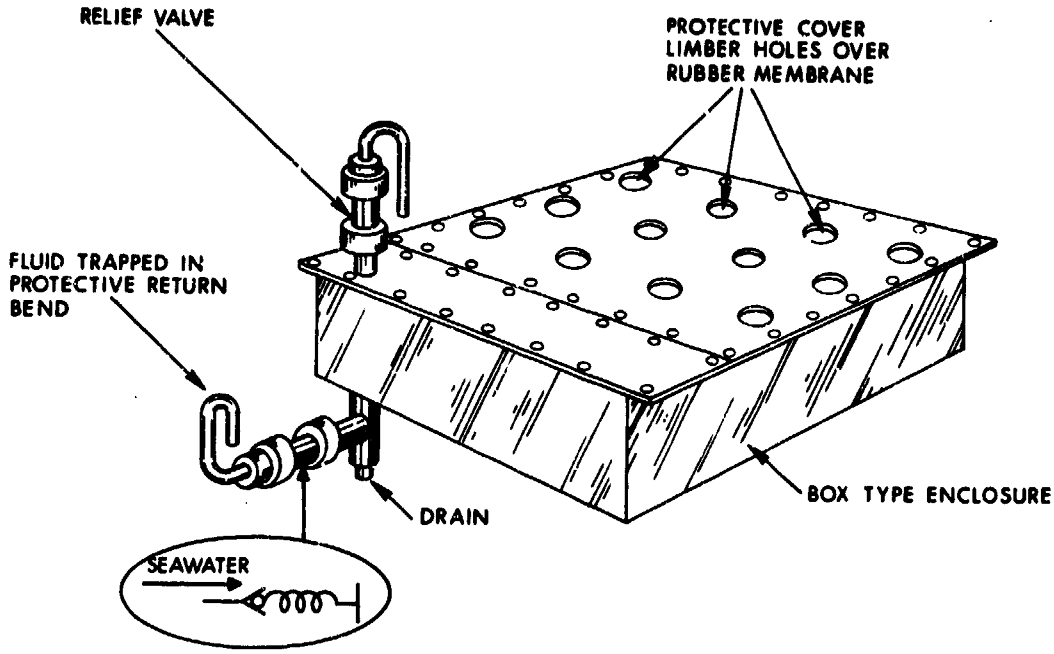
2.3.6 Optimized System Physical Arrangement for Space Utilization, Reliability

A basic, pragmatic approach to the problem of excessive fluid leakage at dynamic machinery seals is to oversize the compensating device. In section 2.6, various physical arrangements are discussed which can enhance seal system-compensating system reliability. Should the fluid volume in a compensating device become depleted, the unbalanced external pressure will rupture elastomeric materials, force sea water through dynamic seals, or implode the equipment enclosure. Therefore, design the compensating system arrangement so that the probable failure mode will be fail-sick* rather than fail-dead.*

In the same manner in which rupture disks and check valves relieve internal overpressure, provision can be made in the enclosure design so that ambient sea water will be admitted as the emergency compensating fluid. This will facilitate a fail-sick operating condition in preference to catastrophic implosion (fail-dead). For a check valve or other device which will inlet sea water, it is desirable to protect the outboard port from the ocean environment by a fluid-filled return bend of tubing (as discussed in section 2.3.2). The sketch below shows an arrangement whereby sea water is admitted to the bottom of the enclosure, so that the remaining compensating fluid will be trapped in the upper portion of the enclosure.

*See Glossary.

Otherwise, sea water which is admitted to higher portions of the enclosure will tend to displace the less dense compensating fluid and equipments will more rapidly become sea-water flooded.

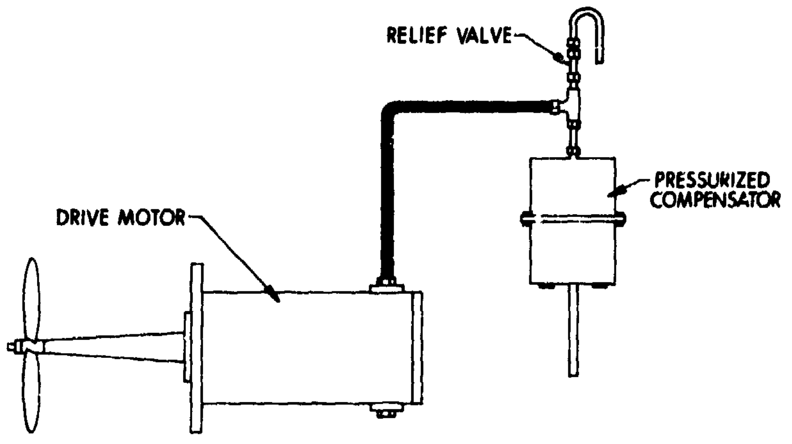


Check Valve Which Admits Sea Water
for Fail-Sick Operation

The advantage of locating an elastomeric tube, bladder, or other unbiased compensator at a point physically lower than the enclosure being compensated is that a small positive fluid pressure is created in the above compensated enclosure (internal p > external p). The positive pressure is relative to the head developed by the difference in specific gravities of the compensating fluid and sea water. In this manner a positive Δp is obtained without using mechanical springs. If leakage occurs, it will tend to be out of, rather than into the enclosure. When this <u>lower</u> compensator arrangement is in air, an unconstrained <u>or un</u>biased elastomeric element or bladder will distend, due to the weight of compensating fluid in the system. This may also create a vacuum in the above fluid-filled enclosure. Moreover, at the air-sea interface, the compensating fluid will experience a pressure transient as sea water contacts and supports the distended elastomer. This negative to positive transient Δp is an opportunity for leakage to occur if system seals become unseated; hence, special design consideration is required.

Therefore, bladders, tubes, or other purely elastomeric compensators should be physically supported and constrained by a housing. Fabric-reinforced elastomeric compensator materials are recommended for this application in that they resist in-air distention or internal pressurization and are more durable in an abrasive sea-water environment (see section 2.2.1).

A negative head can also be developed within an equipment enclosure by locating an unbiased (unpressurized) compensator physically above the system being compensated. This arrangement encourages sea-water intrusion into the enclosure and is therefore not recommended. Similarly, if significant leakage occurs in the compensator, in-rushing sea water may tend to displace the majority of fluid in the system. Another factor is that any gas generated will rise into the line connecting a compensator to the enclosure. A relief valve must be installed at the highest point in the compensating system piping; the compensator must not "trap" gas (see sketch below). When space limitations dictate the use of a higher compensator arrangement, a pressurized compensating device should be used (as shown).



Higher Compensator Arrangement

2.3.7 Fluid Ports/Sampling

Fluid fill/drain and vent ports should be provided at the lowest and highest points, respectively, of each enclosure or fluid-filled cavity in a system. This will facilitate both fluid sampling and maintenance and fill/drain operations between periods of submergence. See the following section for additional coverage.

2.3.8 Miscellaneous Features

Design the compensator for minimum inactive fluid volume, for light weight, and zero sea-water contamination, as well as minimum overall size.

2.4 ELECTRIC DRIVE MOTOR ENCLOSURES

Generally, there are three basic concepts for housing deep submergence electric drive motors. If the design calls for an individual motor case, it is either filled with a fluid and depth/pressure-compensated, or with a-c motors, a sea-water flooded design is feasible. A third option is to combine a motor with other motors and/or components in a fluid-filled enclosure.

2.4.1 Individual Case

General design considerations are as follows:

- Fluid fill/drain and vent ports should be provided at the lowest and highest points, respectively, of each cavity of a fluid-filled motor. This will facilitate vacuum filling and fluid maintenance for the motor and compensating system. A lower cavity or drain sump at the case drain port is desirable to permit a portion of sea-water and particle contamination to settle and collect, rather than to be continuously circulated in the motor fluid. Contaminants in the sump may then be drained off at maintenance intervals.

- Individual motor cases have been designed with depth/pressure compensators built into the antidrive end of the case. In this instance, the cylindrical case is extended to provide a housing for the compensating fluid and spring-loaded diaphragm or piston. This is a desirable design in that the motor and compensator are an integral unit (see section 2.5.2).

- The case material should be thin enough to provide good heat transfer with adequate structural support between the motor case and frame for lifting or manipulating the motor. The deflection and stress on the case due to the cantilevered weight of the motor and compensating fluid must also be considered, especially when a motor is mounted with a single flange. If lifting slings cannot be employed for a certain drive configuration, a lifting eye, cradle, or other attachment should be provided.

- Sea-water flooded a-c motors necessarily have more surface area exposed for heat transfer into the sea water. The rotor is encapsulated or "canned" and the shaft

bearings are sea-water lubricated. For fluid-filled motors, a corrugated case rather than a finned case appears to have more promise for increasing heat transfer. In general, thin organic paints or coatings do not significantly affect heat transfer between the case and sea water.

- Fluid circulating and cooling devices are desirable features to be incorporated with an individual case design. With both a-c and d-c motors, where high localized heating or "hot spots" occur, circulation of the compensating fluid will aid heat transfer and provide a more uniform temperature distribution throughout the case.

- There is a need for filtering devices in fluid-filled motors to prolong life of bearings and wear surfaces. Particle contamination may be continuously circulated in an individual case. Especially with d-c motors, the problem of carbon brush wear contamination is enhanced by the limited fluid volume available to dilute the carbon. Bearing wear may increase and the dielectric strength of the fluid will decrease as carbon contamination ensues. Carbon buildup in a fluid also increases the possibility of d-c arcing and grounding. Another additive effect of carbon contamination is that d-c commutation tends to degenerate.

- It is most feasible to employ a sea-water floodable a-c motor as a direct-drive unit, whereas a fluid-filled motor may be either direct drive or coupled with a speed reducer. In the latter cases, with an outboard shaft seal, there is the possibility of fluid leakage or intrusion of sea-water and silt contamination. Metal wear particles will also enter the fluid from the speed reducer and motor. Both thermal compatibility and the transfer of contaminants from a speed reducer to a motor or the converse must be considered, especially when a common compensating fluid is used.

2.4.2 Combined with Other Components in a Housing, Compartment, Etc.

The use of fluid-filled, pressure-compensated enclosures to protect sensitive components from the sea-water environment is basic to deep submergence design. Motor-pump units for hydraulic power systems, variable ballast, or trim systems are typically installed in this manner. Hydraulic and electrical components and one or more electric drive systems may be mounted within one compartment. This arrangement is not only convenient, but increased reliability and safety through system redundancy is possible. Electrical and hydraulic feedthroughs are reduced.

The general design considerations are as follows:

- As mentioned previously, a combined motor and component arrangement will reduce the number of outboard electrical cables and hydraulic lines. Bulkhead penetrations are minimized; fewer fluid to sea-water seals are required. Since various components are not separately fluid pressure compensated, there is a saving in space and weight.

- In contrast to an individual motor case, the larger fluid volume of this type of compartment will allow greater dilution of particle contaminants and sea-water leakage. For d-c systems, the dilution of carbon clinkers and brush wear debris will tend to preserve fluid dielectric strength. At maintenance intervals, contaminants which have settled to the lowest portions of the compartment can be drained off. It is also necessary to provide a vent port at the highest part of the compartment and a fill/drain port at the lowest point for fluid filling and maintenance procedures.

- The large volume of fluid and increased surface area of this enclosure provide good heat-transfer characteristics. With moderate operating temperatures, bearing life and d-c brush life are enhanced.

- Contactors in a motor starter assembly (mounted on or near the motor) may be left uncovered to the circulation of surrounding fluid to discourage the buildup of "clinkers" in the contacts. Fluid circulation from operation of the motor has an effect of "washing down" contactors and dispersing carbon particles.

- A preferred design for a hydraulic pump-drive arrangement is that the hydraulic fluid system be a <u>closed</u> system so that the machinery compensating fluid <u>will not be</u> the system reservoir. As a minimum design requirement, the hydraulic power transmission fluid should be separate from compensating fluid for the drive motor and sensitive electrical components (see section 2.7.3). Otherwise, all components in the compartment will be exposed to contaminants in the system fluid, thus defeating the reliability aspect of this configuration. This is especially critical for electrical equipment. A compensating fluid should be chosen which has good materials compatibility, adequate electrical properties, corrosion protection, and yet will provide satisfactory lubrication for electric motors. Contaminants tend to deteriorate most critical fluid properties.

- Generally, submersible motor controllers have been designed as separate packages from drive motors.

Although the use of a separate controller entails additional cabling and another pressure-compensated housing, it has been the more feasible approach for the following reasons:

- For any combination of motor and controller, drive pulsations, vibration, and stray magnetic fields from the motor may misalign controller circuitry and impair performance, control, and reliability.

- Temperature-compensating circuitry in the controller logic would experience considerable fluid temperature transients as load changes and reversing operations of the motor occur.

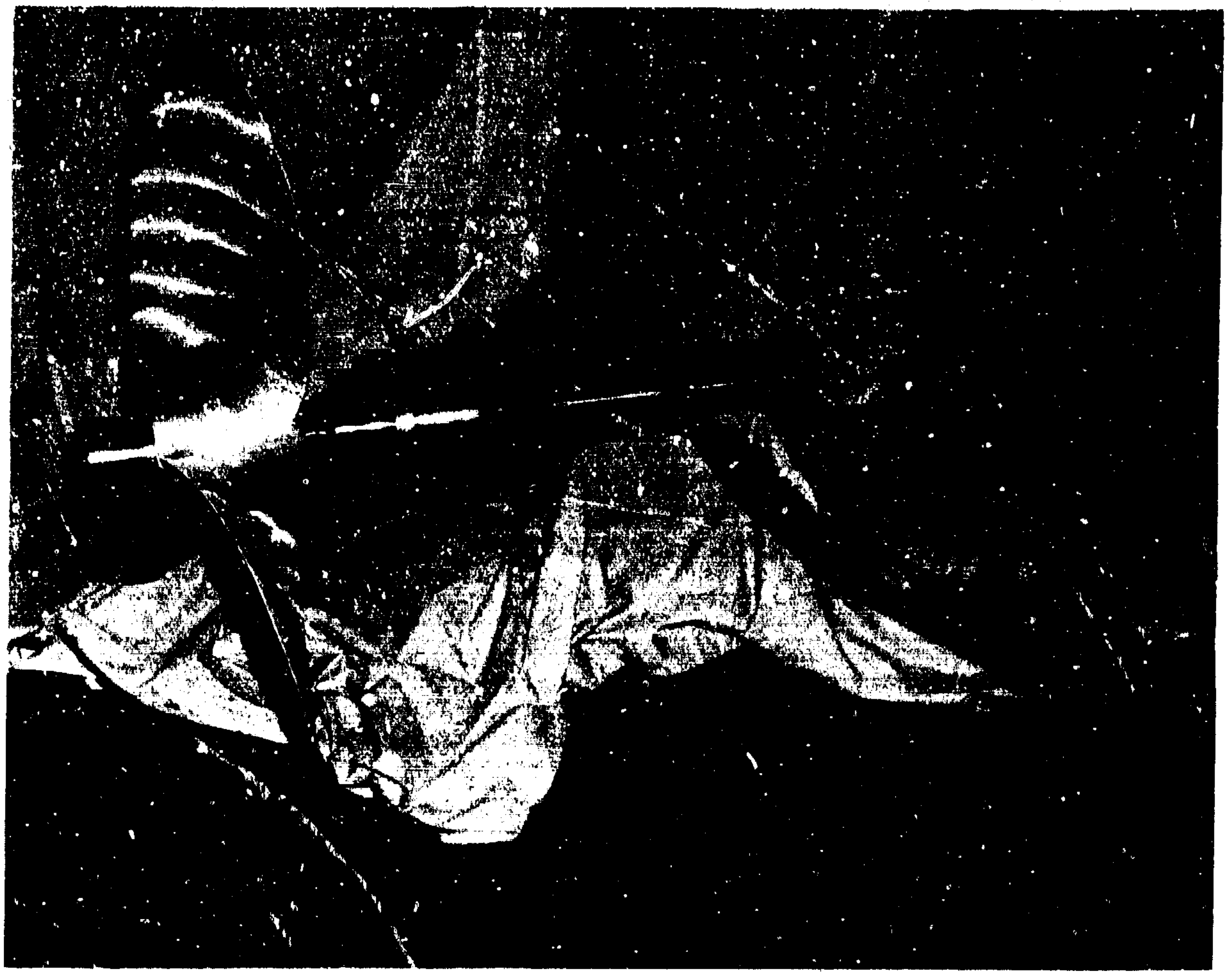
- Carbon contamination in the fluid may affect controller circuitry and encourage grounding or shorting.

- When an application calls for a combined motor, speed reducer, and controller package, the motor to controller interface must be carefully engineered to minimize interference with the controller and insure reliability. Overall size and weight of this configuration are somewhat prohibitive, especially for units higher than 10 hp. Other potential problems are high cantilevered weight and mounting considerations for adequate support.

1 - Speed Reducer

2 - Motcr

3 - "Wet" Electronic Motor Controller



Developmental Submersible Electric
Propulsion System

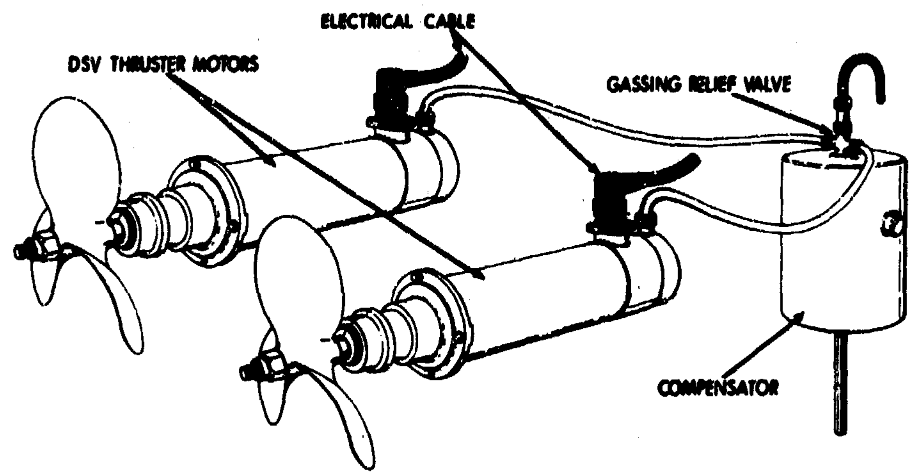
2.5 PHYSICAL ARRANGEMENTS FOR COMPENSATING SYSTEMS AS APPLIED TO SUBMERSIBLE ELECTRIC DRIVE MOTORS AND SPEED REDUCERS

2.5.1 <u>Central Compensator</u>

- An arrangement whereby compensating fluid is communicated to two or more separate motors (housings, components, etc) from one external compensator. It is

feasible to fluid compensate an entire drive system in this manner.

• One application includes thruster motors on a DSV that are connected to a common, remote compensator. The compensator should be physically located below the motors, or at the same level as the motors, depending upon space limitations, drive system design, and maintenance philosophy. Protective features should be provided in the design and installation of the compensating system to accommodate fluid gassing and sea-water leakage. These considerations were described in the previous sections.



Central Compensator

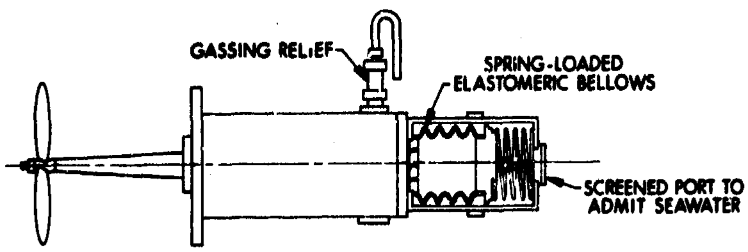
• The use of a central compensator affords simplicity and possible savings of weight and space. Since the compensator is somewhat remote from the operating components, the central compensator fluid volume is not subjected to heat generated in the compensating fluid within each housing. The total fluid expansion is then less than for an internal compensating system.

• System reliability is decreased from the standpoint that a considerable loss of fluid from any part of a central compensating system would result in a probable malfunction or failure of all components being compensated.

2.5.2 Individual Compensator

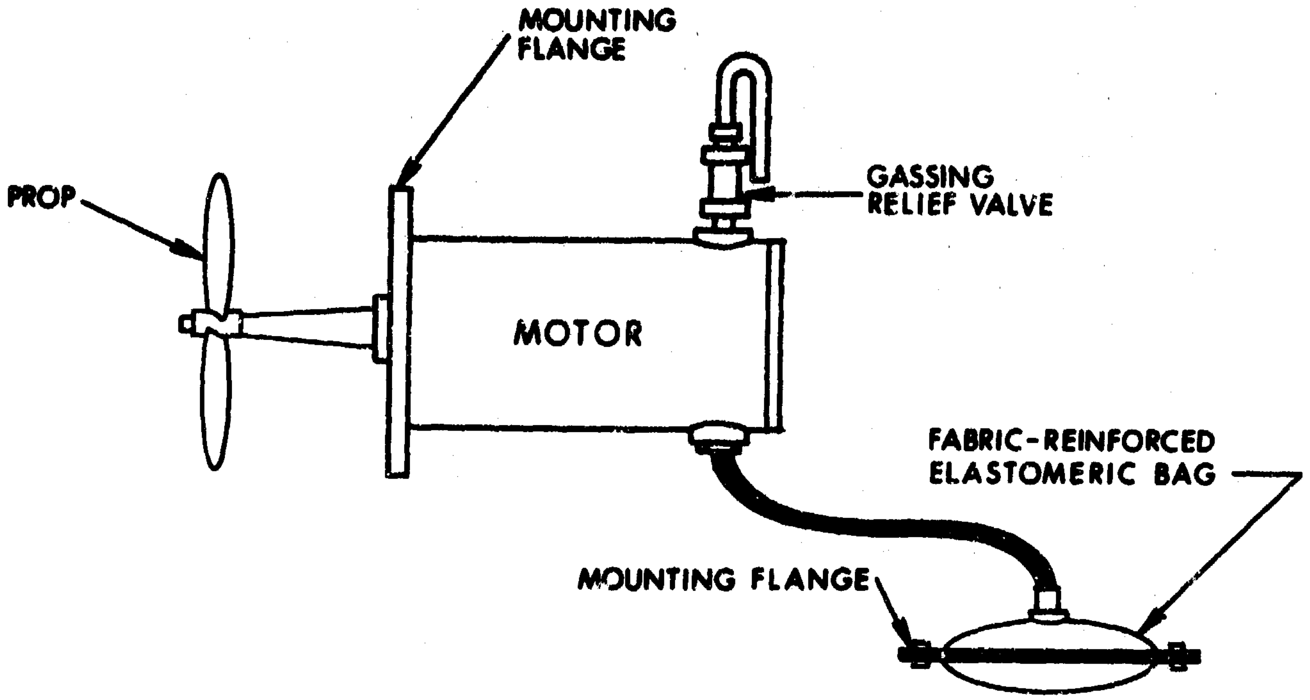
• Internal. A configuration whereby the compensating device is mounted integrally within or on an equipment housing. A motor with (or without) a speed reducer, or a controller, can be internally compensated

each as a separate unit, combined separately, or as one complete physical assembly. An internal compensator can be mounted as an integral extension of the equipment housing, such as a spring-loaded bellows or rolling diaphragm mounted into the antidrive end of a motor case. Another concept employs a circular elastomeric tube, where sea water is admitted to the inside of the tube to equalize the compensating fluid to depth pressure. Since an internal tube cannot be readily examined and tested for leaks, the condition and reliability of the compensating system can only be determined by disassembly of the housing at maintenance intervals. Therefore, an internal-tube compensating device is not recommended.



Internal Compensator

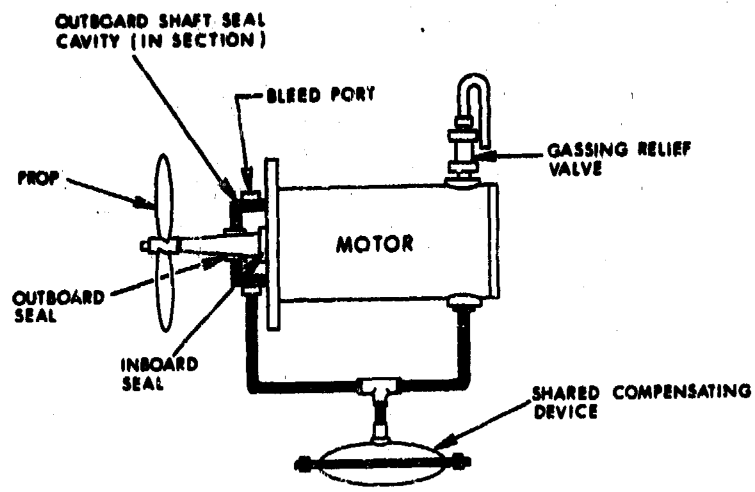
• External. The compensating device is mounted outside the equipment housing as a separate or even remote compensator. Tubing, pipe, or hose is used to communicate compensating fluid to the equipment. As mentioned in the central compensator discussion, drive system heat transferred into a separate, external compensator is less than for an internal compensator; thus, less total fluid expansion will take place. A separate, external compensator is more accessible for maintenance, disassembly, and modifications. An external compensating scheme is also versatile and readily adaptable for various operational requirements of a drive system. Virtually all types of compensating devices may be used.



External Compensator

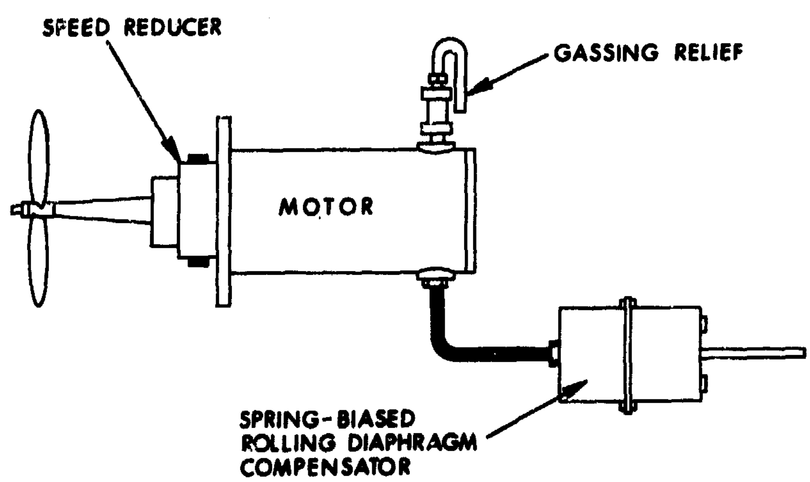
- Various schemes for compensating an individual drive system

  - Shared. A shared compensator is similar to a central compensator in that fluid is communicated to various compartments or cavities, but differs in that only <u>one</u> combined equipment assembly is involved. The compensating device is external. Fluid volumes in adjoining housings are essentially separate, even when a shaft seal is employed at the interface. A shared compensator affords an equalization of internal fluid pressure rather than a differential pressure across equipment interfaces. This arrangement either precludes or minimizes fluid mixing and transfer of contaminants between adjoining housings, although a common compensating fluid is used.



Shared Compensator

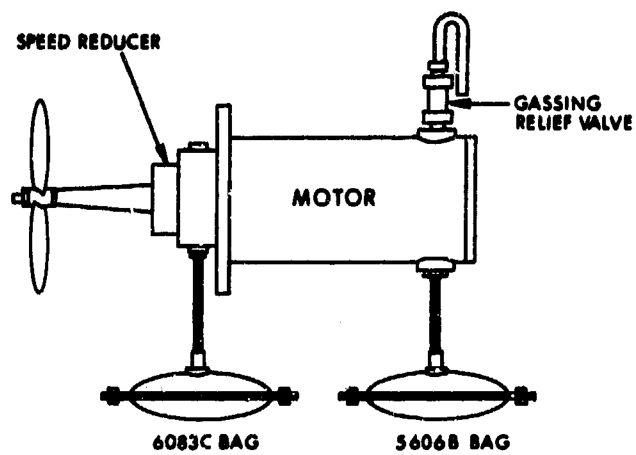
Common. This is where one compensator is employed to compensate two or more interconnected compartments or housings, such as a motor and speed reducer. A common compensator may be either external or internal to the drive system housing. Since fluid is free to circulate, contaminants may be transferred from one compartment to another.



Common Compensator

Separate. If the above arrangement was modified so that an interface with a shaft seal was provided between the motor and speed reducer, the transfer of fluid and contaminants would be minimized. If a shared compensating arrangement is not used, two separate compensators may be employed. For the motor, fluid

electrical properties are often critical and can be better preserved by separate compensation. Another option is that different fluids may be used in each compartment. Instead of a compromise of fluid properties, an optimum fluid may be chosen for the specific requirements of the motor and likewise for the speed reducer. The sketch below shows separate, external compensators for one drive assembly. For example, MIL-H-6083C fluid may be used for the speed-reducer compensating fluid, while MIL-H-5606B fluid may be used in the motor.



Separate Compensators

2.5.3 Compensated Enclosure for Combined Equipment

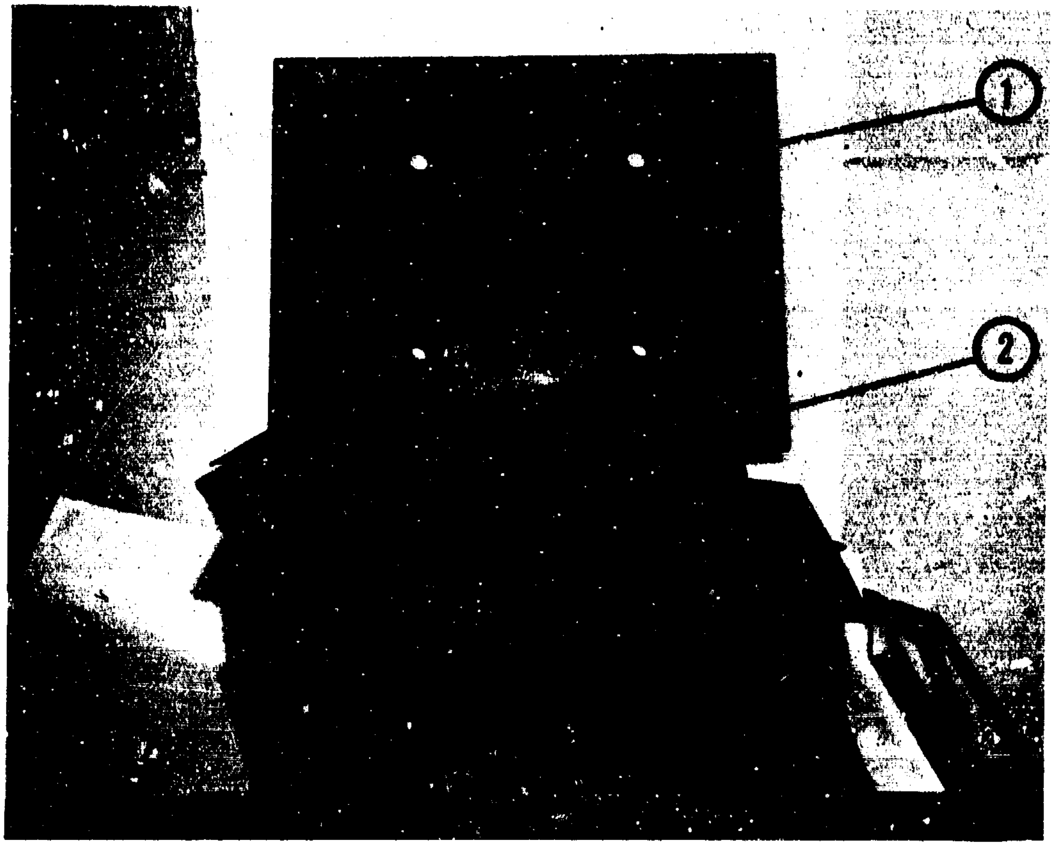
This is basically a hard structure with one wall as a flexible membrane or diaphragm so that one or more drive motors and various hydraulic or electrical components are housed with one protective fluid environment. See sections 2.4.2 and 2.7.3 for details and considerations for compensated enclosures with combined equipment. If the walls of a box-type enclosure are made rigid, a spring-biased diaphragm or piston may be employed to slightly pressurize the compensating fluid above ambient sea-water pressure. Leakage will tend to be from the compensating fluid into the sea water rather than the less desirable alternative.

A fluid depth/pressure-compensated system may experience a fluid transient shock at the air-sea interface. For this type of enclosure, the pressure transient can be severe, due to the large exposed surface area of the flexible diaphragm and weight of compensating fluid which is constrained by the diaphragm. Thus, special design consideration is required to avoid transient leakage into or out of the enclosure and possible rupture

of the flexible diaphragm. It is recommended that the diaphragm or membrane be installed to the top, rather than the side or bottom, of the enclosure so that at no time will the elastomer distend due to the weight of the compensating fluid (see section 2.7.2).

1 - Protective Cover

2 - Flexible Membrane



Combined Equipment Enclosure
for Hydraulic Power System

1 - Solenoid-Operated Valves

2 - Hydraulic Lines

3 - Pump and Motor

4 - Self-Pressurizing Reservoir

5 - Stuffing Tubes for Electrical Cabling

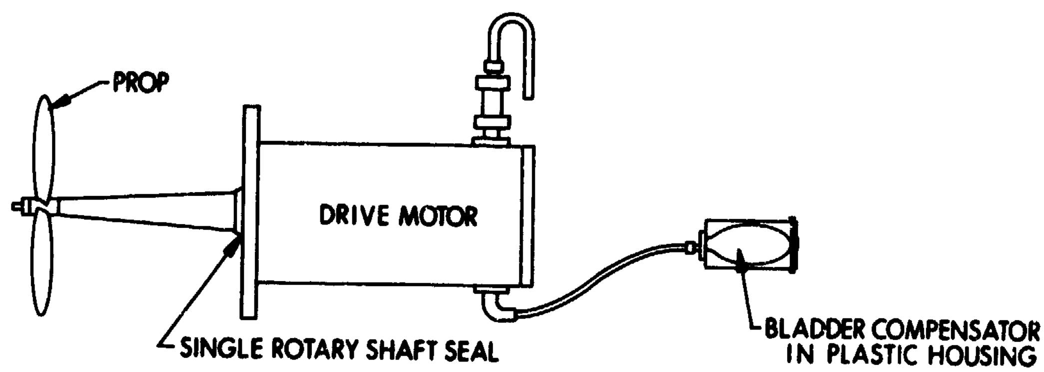
Combined Equipment Enclosure
for Hydraulic Power System

2.6 SIMPLE, REDUNDANT, CASCADED, AND PRESSURE-CASCADED COMPENSATION

For operation near the bottom, drive system seals at the machinery-sea interface will be subjected to sand, silt, marine growth, and various other abrasives. The use of protective slingers or centrifugal separators on rotating shafts will minimize the amount and size of

abrasives which reach shaft seals. Internal particle contaminants from thermal fluid decomposition and bearing and brush wear may also be present in the compensating fluid.

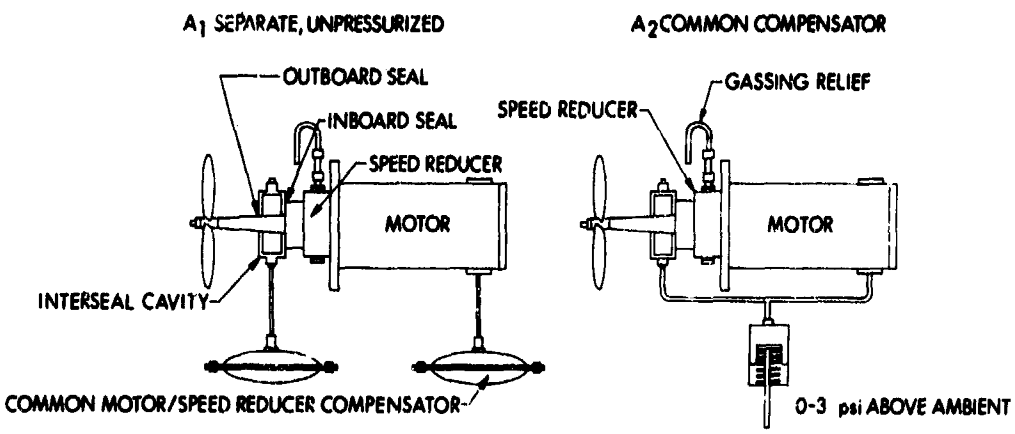
A simple compensating configuration may be either unpressurized or pressurized, but the drive machinery fluid will be separated from the ambient sea-water environment by a single shaft seal assembly (see sketch below). Ease of fluid filling, draining, and general maintenance are advantages for a simple arrangement. This approach is used for drive systems having relatively short submergence time; hence, frequent inspection and maintenance.



Simple Compensation

Redundant, cascaded, and pressure-cascaded compensating systems are used for additional fluid protection of drive motors and/or speed reducers which are sensitive to sea-water leakage contaminants. A redundant compensating arrangement provides an intermediate, fluid-compensated cavity and a dual seal system at the machinery-seal interface. An outboard shaft seal separates the fluid-filled interseal cavity from the sea while an inboard shaft seal separates the drive machinery fluid from the interseal cavity (see sketch A). In a redundant, dual-seal system there is no pressure drop across the seals (by definition). Note that the machinery housing and interseal cavity are always maintained at the same pressure, whether two separate unpressurized (A_1) or one pressurized or unpressurized compensator (A_2) is used.

Sketch A



Redundant Compensation

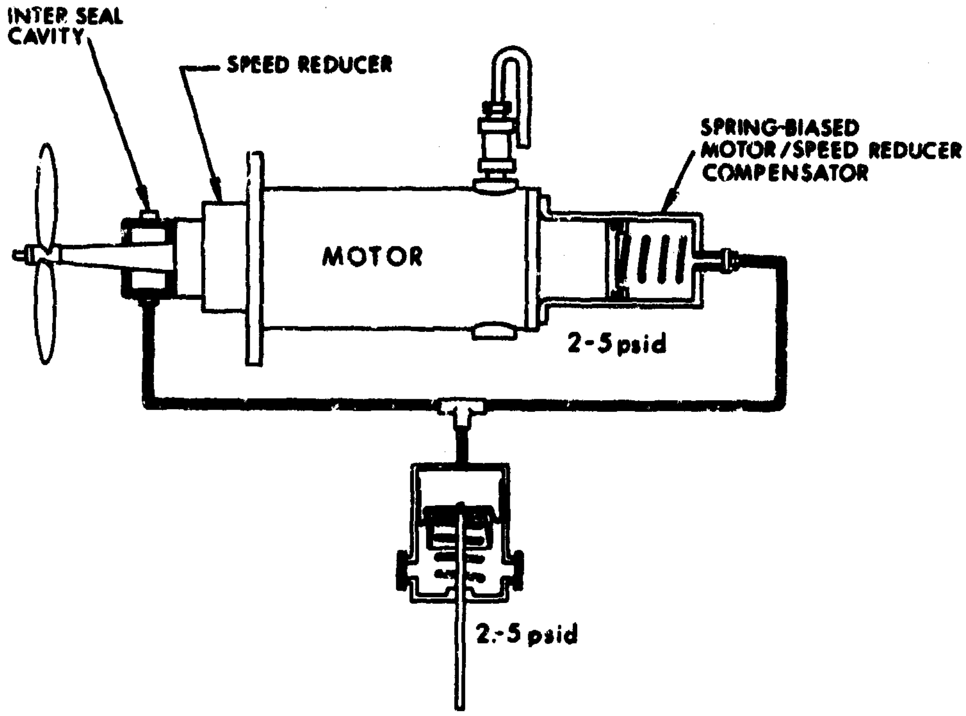
In a dual-seal system, note that ports must be provided at the top and the bottom of the interseal cavity. This will facilitate fluid filling, draining, and the attachment of a compensator. A nonemulsifying interseal fluid may be used to allow periodic draining of sea water and abrasive particles which settle to the bottom of the cavity.

If the separate compensators in sketch A_1 are spring-biased and cascaded to provide a differential pressure across the shaft seals, the compensation will be pressure-cascaded. For instance, the interseal cavity may be pressurized 2-5 psid above ambient sea pressure and the machinery housing is then maintained at 2-5 psid above the interseal cavity (sketch B_1) or 4-10 psid above ambient.

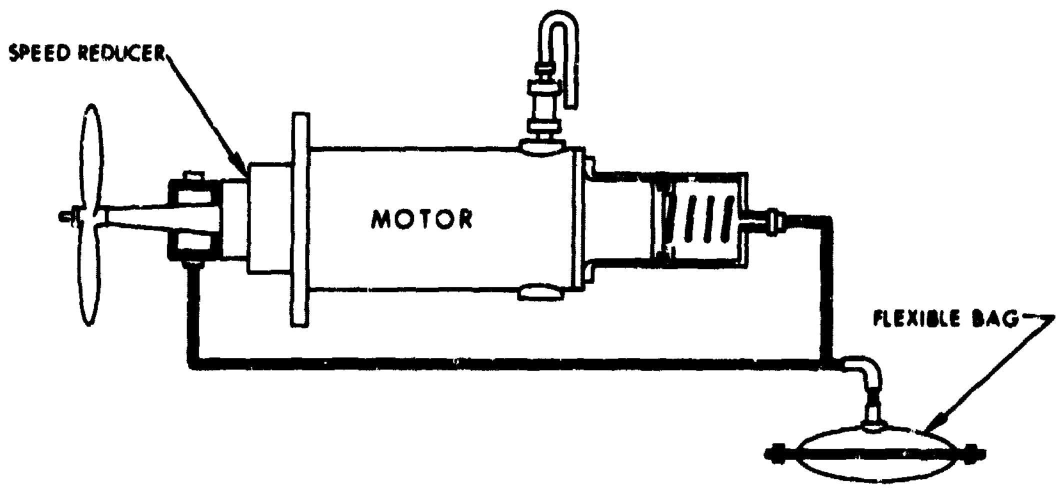
Note in sketch B_2 that the interseal cavity is unpressurized. Also, the outboard side of the motor-speed reducer compensator is cascaded by this unpressurized, shared, external compensator. Rather than admit sea water and particle contaminants to the internal housing, spring, piston, or elastomer, cascading the compensation provides a protective fluid environment for normally exposed parts.

Sketch B

B_1 Pressure-Cascaded



B_2 Cascaded



In either case, the design philosophy for a dual-seal system is that the rate and amount of sea-water intrusion into the speed reducer or motor fluid will be reduced. A longer leakage path is provided between the sea and machinery. As the outboard seal is exposed to abrasives in the sea water, only particles less than the seal clearance (up to 5 microns) can enter the interseal fluid. The filtering action of the outboard seal provides abrasive protection for the inboard seal. The outboard seal may feasibly incur significant wear and fluid leakage so that the interseal compensator volume is depleted. Thus, the interseal cavity should be sea-water floodable as a worst condition; otherwise, both seals will fail catastrophically.

The main advantage in redundant or cascaded arrangements is the "double dilution" of sea water and particle contaminants. Rather than pressure-cascading the drive assembly, there may be more merit in slightly pressurizing the drive machinery fluid alone and cascading the compensation (see sketch B_2), for the following reasons:

- Outboard seal abrasive wear may be excessive so that significant interseal fluid leakage into the sea water can occur.

- Pressure-cascading the interseal cavity will increase the fluid leakage rate so as to deplete the compensator. The interseal cavity would become sea-water flooded as a "fail-sick" operating condition, thus defeating the purpose of a dual-seal system.

- In an unpressurized interseal cavity, whether cascaded or separately compensated, although some local sea-water-fluid mixing may occur, the cavity will continue to function as a contaminant-diluting fluid barrier for the inboard seal.

- Slightly pressurizing the drive machinery fluid will discourage the intrusion of contaminants through the inboard seal and will offset any transient pressure reversals.

The use of a dual-seal arrangement results in more rotational losses and complexity for a given drive system. A decision must be made on whether the complication of an additional seal is worth the protection it affords and the improvement in shaft seal wear and leakage.

2.7 HYDRAULIC SYSTEM COMPENSATION

2.7.1 Description

Electrohydraulic power systems continue to serve as the primary energy transfer systems in hydrospace

applications. By oil-filling and pressure-compensating an enclosure which contains the drive motor, pump, valving, and associated hardware, the hydraulic system pressure is continually referenced to a "floating" base line pressure - that of the ambient ocean. In other words, with an external pressure of 9000 psi (20,000-foot depth), hydraulic components in a nominal 3000 psi system will see a total internal pressure of 12000 psi while the maximum differential pressure in the system will be 3000 psi. Self-contained hydraulic systems have been designed for shallow submergence applications, such as hydraulic power units for diver tools. The precharged accumulators employed in these shallow-depth systems pressurize the system fluid against the maximum ambient sea-water pressure. However, with increasing depth, the weight penalty of the required accumulator housings and other pressure-resistant hardware becomes excessive.

Within a typical hydraulic package, fluid is manifolded and directed to various functions through solenoid-operated transfer valves. Since the main hydraulic system is usually considered critical to vehicle safety, two identical main hydraulic systems (or at least motor-pump units) will provide redundancy, extra emergency power, and thus, reliability and safety. "It has been estimated that as much as 70% of all hydraulic problems may be traced directly to the condition of the fluid."[1] Diligent fluid sampling and reconditioning are major factors in obtaining reliable performance. Contamination from an integral, hydraulically powered mercury trim system, or manipulator system could be quite detrimental to main hydraulic system components. Therefore, it is desirable to design separate, simplified hydraulic power packages to serve individual subsystems and preclude the transfer of contaminants.

DSV hydraulic systems have been designed to supply fluid power for the following subsystems or functions:

- Stern propulsion hydraulic motor.
- Stern propeller shroud actuation (steer).
- Thruster hydraulic motors.
- Variable ballast system.
- Trim system.
- Manipulator(s), coring devices.
- Selective weight release.
- Actuators and winches.

Hydraulic circuitry must be scrutinized to ensure that no components or circuit elements have trapped or uncompensated volumes. With changes in depth and fluid pressure, all sections of the hydraulic system must be relieved to the system reservoir or otherwise individually pressure compensated. For instance, during vehicle ascent and descent, inactive control valves may be placed in an open-center position so as to relieve trapped system fluid.

In the main hydraulic system and associated subsystems, the use of suitable valving to provide pressure relief in the event of high-pressure loading or other unusual operating conditions is advised. Relief valve locations, pressure settings, filter locations, degree of filtration, and other significant hydraulic system requirements may be found in the SAE "Proposed HIR for Hydraulics Systems for Deep Submergence Vehicles," 1 March 1971, SAE A-6A Subcommittee, Underwater Fluid Power & Control Systems Panel.[38]

2.7.2 Compensating Device

Usually a box-type enclosure is designed to house the hydraulic pump drive assemblies combined with servo valves, filters, electrical components, etc. The enclosure will not have any rotating shaft penetrations, only hydraulic and electrical feedthroughs.

To compensate the enclosure, one approach is to install a diaphragm or flexible membrane to seal one enclosure wall. To reiterate, it is recommended that unbiased elastomeric material be installed at the top, rather than the side or bottom, of the enclosure to avoid elastomeric distention due to the in-air weight of the restrained compensating fluid. If the enclosure walls are stiffened or made rigid, a spring-biased compensating device may be employed; the enclosure cover is gasket-sealed.

Box-type enclosures are often of welded construction with flanged, gasket-sealed openings. Both 6061-T6 aluminum with a hard anodized finish coating and 316L stainless steel have been used. For protective membrane covers, PVC and prelaminated fiber glass appear to be as good or better than the above enclosure materials. Not only are these materials light, strong, and easy to fabricate, but they also do not deteriorate in the high-pressure ocean environment.

Protective covers for membrane-sealed enclosure openings should be fabricated with numerous small diameter "limber holes" over the surface area of the thin, flexible

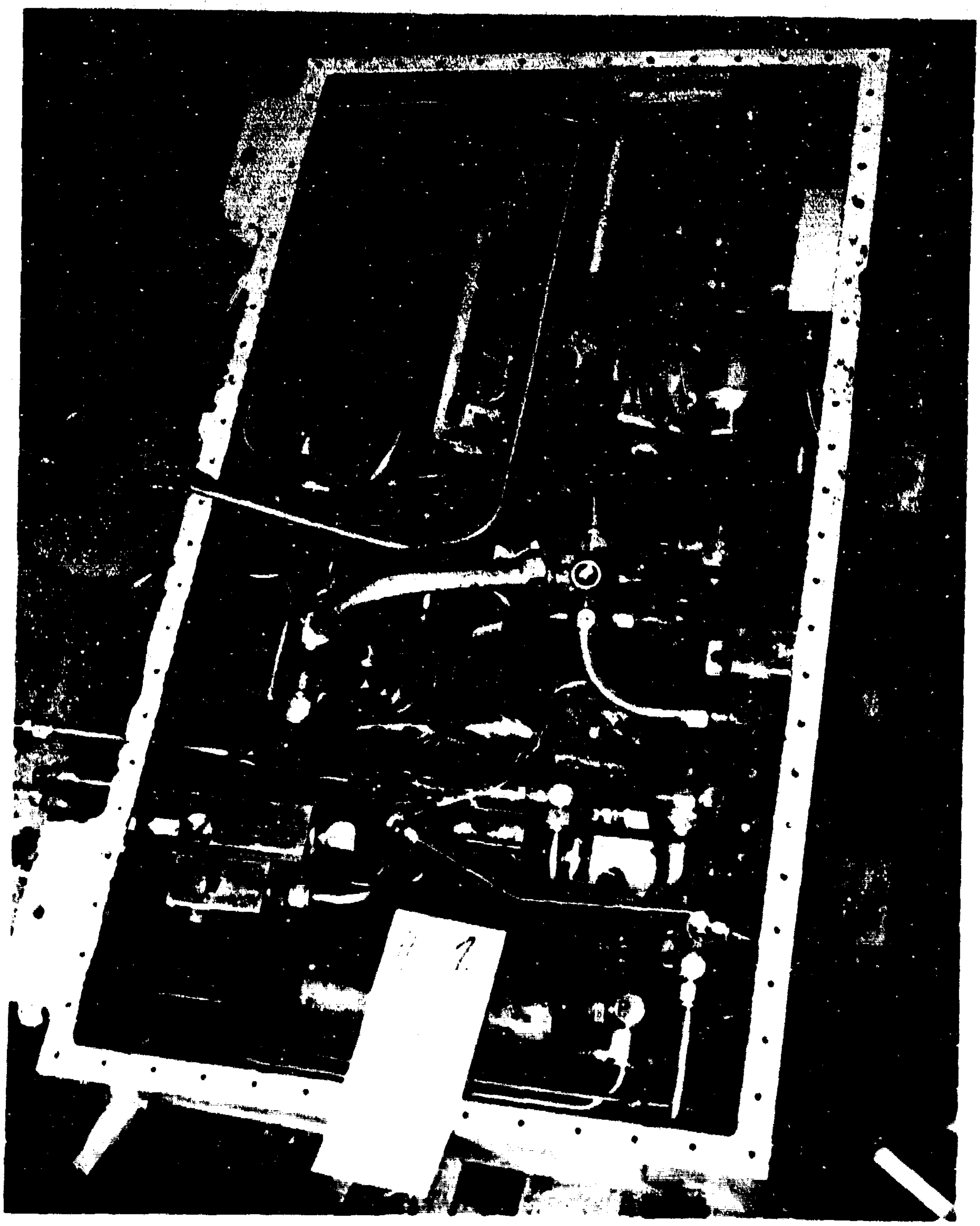
material. The limber holes will communicate sea water to the membrane, enhance cover flexibility, provide environmental protection, and prevent membrane blowout. To aid the compensation, the resilient flexure of the membrane cover will tend to maintain an initial enclosure overpressurization.

Fluid ports and/or quick disconnects should be provided at accessible locations to facilitate sampling, fill/drain operations, and in-place fluid reconditioning by means of an external purification unit. The top portion of the enclosure or associated compensator tubing should include a low-pressure relief check valve to vent overpressure directly to ambient sea water. Nylon check valves, set for 1 psi relief pressure, have been used for this enclosure application. A return bend protective device is recommended for the sea-water-exposed relief-valve outlet.

In order for operating personnel to be warned of sea-water intrusion, a sea-water leakage detector should be installed in the bottom of the enclosure(s). Where the combined equipment enclosure design does not have a sloping bottom or otherwise facilitate a sump, a detector should be installed in each bottom corner and the detectors should be wired in parallel.

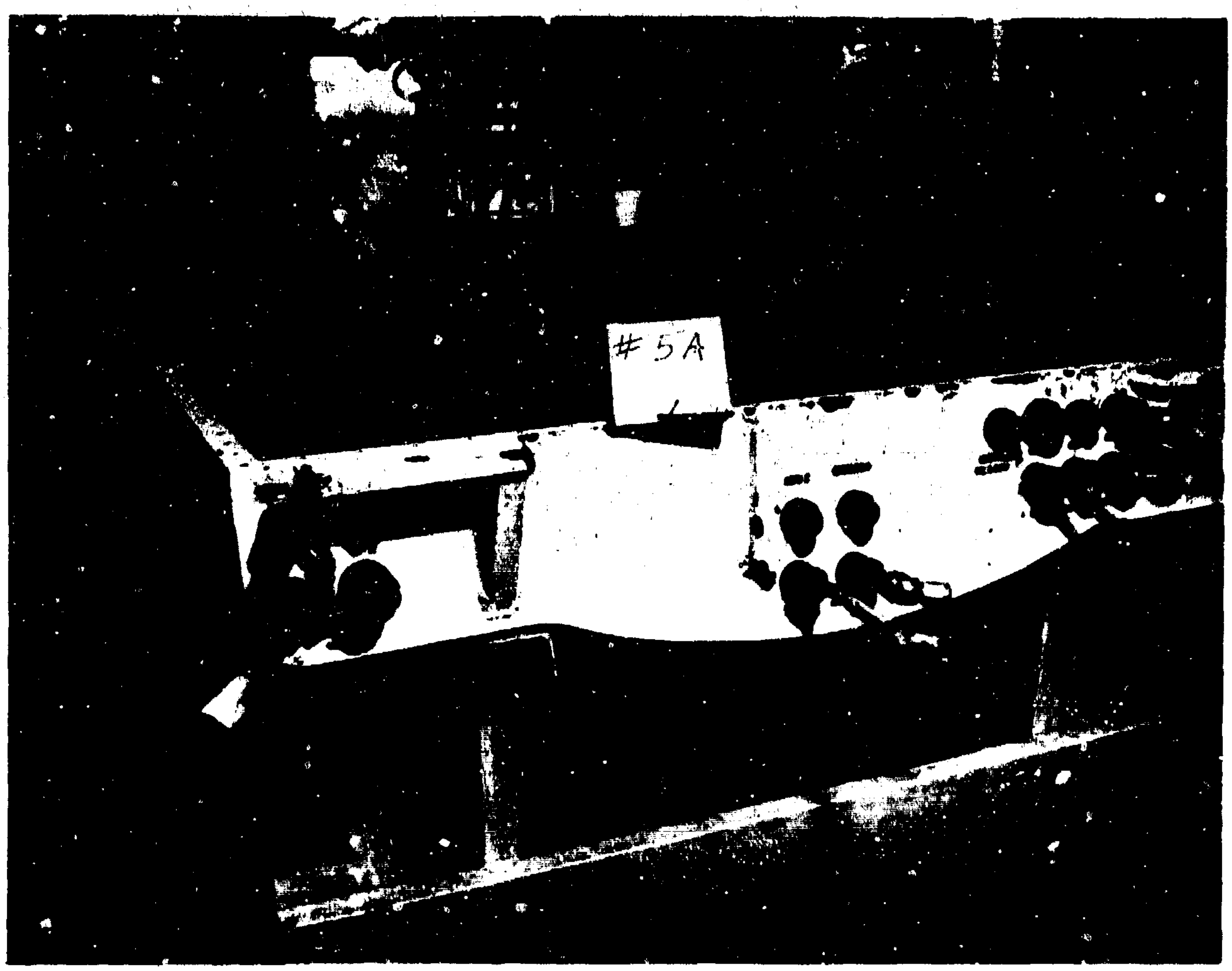
The use of solenoid-operated hydraulic valves which have mechanical detents or contacts should be avoided. As contacts make and break electrically, contact fouling is a common problem. It is desirable to mount d-c relays, contacts, etc in well-exposed locations to permit compensating fluid to "wash down" and dilute breakdown products from electrical arcing.

Item (a) - Top View



DSV "ALVIN" Hydraulic Propulsion Box
(Membrane and Ported Cover Plate Removed)

Item (b) - Side View



Note: The bottom of this enclosure is exposed at and contoured to hydrodynamic vehicle fairings, which enhances heat rejection and provides accessibility for maintenance.

2.7.3 Enclosure Arrangement Details

Three basic arrangements for hydraulic system enclosures have been used on DSV's. These are discussed in ascending order of desirability.

- Open System - an arrangement whereby all combined equipment in the enclosure is immersed in the system fluid. The enclosure compensating fluid and the hydraulic system fluid are one and the same. Fluid selection will likely be a compromise situation in order to satisfy various mechanical, hydraulic, and electrical fluid requirements. Thus, the enclosure also serves as the system reservoir and hydraulic return lines dump directly into the enclosure fluid volume. All hardware is exposed to contamination which may enter or be generated within the entire hydraulic system. Obviously this is the poorest compensating approach to insure good equipment life and reliability and therefore is not recommended for systems which include electrical and electronic equipment.

- Separate Compartments - as an improvement over the open system layout, the enclosure is designed to have two separate, individually pressure-compensated compartments. One compartment houses the electric pump drive motor(s) and sensitive electrical components. A compensating fluid may then be selected which best suits motor and electrical system requirements. The adjoining compartment remains the hydraulic reservoir and houses the hydraulic pump(s), control valves, and associated components. Accordingly, an optimum hydraulic fluid may be selected.

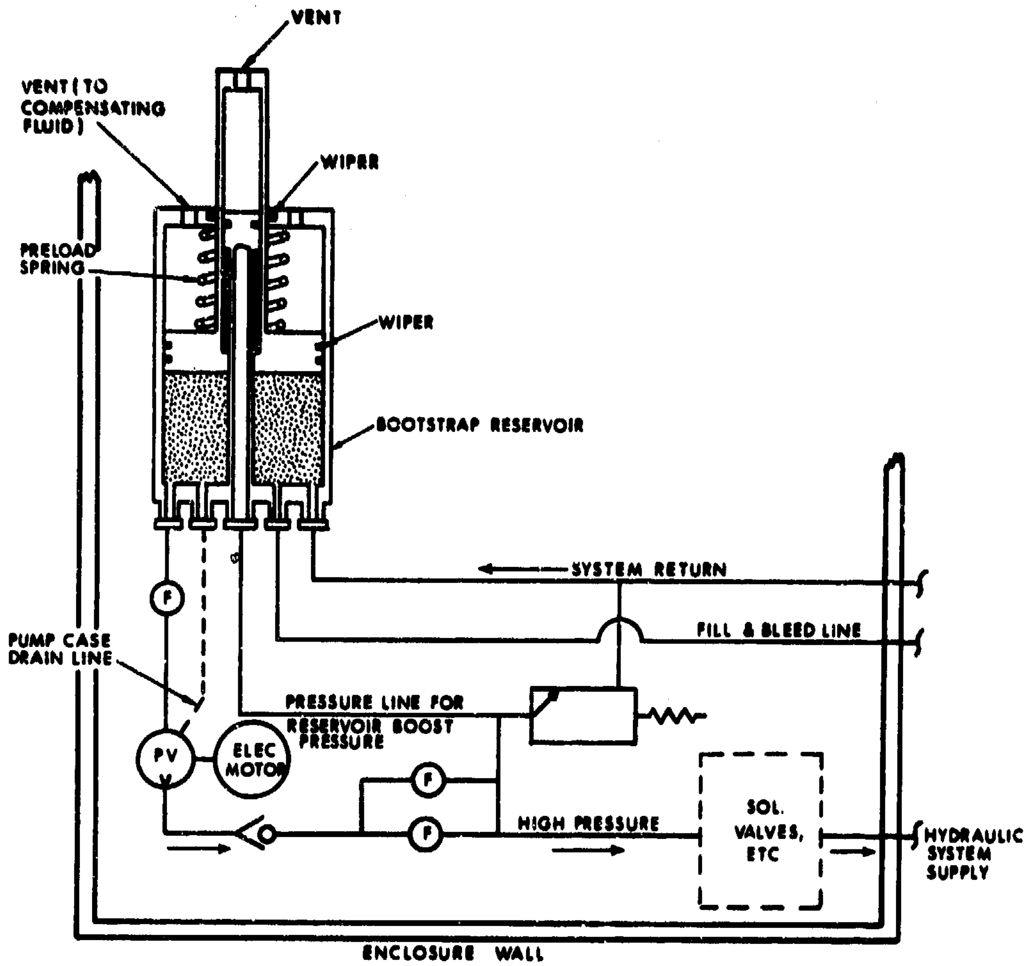
However, servos, solenoids, and close-tolerance hardware will remain vulnerable to virtually all types of harmful contaminants. At the motor/pump interface, a mechanical shaft seal will be provided. Care must be exercised in uniformly pressure-compensating each compartment; otherwise, a pressure differential may encourage seal "pumping." A bidirectional rotating shaft seal will minimize and perhaps preclude the interchange of fluids and contaminants between the compartments. But the integrity of this compensating concept is largely dependent upon the design, installation, and performance of the shaft seal.

- Closed System - this is a design in which the fluid compensating the entire combined equipment enclosure is separate from the hydraulic system fluid. The hydraulic system employs a bootstrap, or self-pressurizing, reservoir which is immersed in and compensated by the enclosure fluid. In this arrangement (see schematic below), system high pressure enters the central bootstrap reservoir port and reacts against high- and low-pressure pistons creating

the desired low pressure. Notice that both cylinders are vented to the enclosure fluid. The pump inlet pressure boost achieved is a function of the piston area ratios (say, 1/60th of outlet pressure). There is a possibility of pump cavitation for an open hydraulic system or with separate compartments, even if the enclosure fluid is slightly pressurized. For DSV hydraulic propulsion systems, maximum power is often required during launch/recovery operations, particularly for positioning the submersible in heavy sea states. Cavitation is most proncunced under these high-load conditions with low ambient pressure, high flow rates, and high fluid temperatures. However, the use of the bootstrap principle minimizes cavitation problems.

This closed system concept essentially eliminates contamination of the enclosure-compensating fluid by the hydraulic system and thus readily lends itself to the use of d-c machinery systems and fluid-filled, "self-hosing" electrical cabling (see section 2.8.2). Optimum compensating and hydraulic fluids may be selected for system requirements. Another benefit is that hydraulic machinery may be immersed in compensating fluid which is different in color from system fluid, to aid in leak detection. For example, when the enclosure cover or membrane is removed, external leakage of red hydraulic MIL-H-5606B will readily discolor the clear Hoover(TM) Submersible 2 compensating fluid.

(TM)Trade names marked with this symbol are proprietary to the manufacturer.



Closed Hydraulic System

2.7.4 System Design and Arrangement

For increased reliability through redundancy, DSV's have been designed with two separate hydraulic systems; each system may even have its own battery power pack. In another design, the main hydraulic system supplies power for propulsion and steer while a second hydraulic system normally powers the variable ballast system.

These two systems should be separated electrically, and separate fluid-filled, pressure-compensated electrical distribution systems should be provided. Perhaps the only mechanical connection between systems would be a four-way valve. If the main electrical distribution box were flooded with sea water, the variable ballast pump, with separate electrical power, would function as the main hydraulic system pump. The VB hydraulic system can also be used to supply extra lift power or speed in an emergency or in case of a failure in the main hydraulics.

Likewise, with a failure of the VB motor or pump, or if faster pumping was needed, the main hydraulic system pump may be cross-connected.

2.8 ELECTRICAL/ELECTRONIC COMPONENT COMPENSATION

2.8.1 Description

Outboard electrical systems have been one of the most critical and worrisome problem areas in the development of deep submergence vehicles. During a dive, if an electrical enclosure or component becomes flooded, it is difficult to determine the cause and exact location of the initial failure. Moreover, failure reports of operational vehicles indicate that outboard electrical cables and connectors are frequently responsible for the flooding of outboard electrical systems.

For a manned DSV, control and display equipment is located, by definition, within the personnel capsule; a multitude of outboard electrical components must be packaged and interfaced to lock out the corrosive and short-circuiting effects of sea water. Components such as lights, sensors, still and TV cameras, hydrophones, and various transducers are usually housed in a hard shell-type enclosure. For operational depths up to 2000 feet, encapsulation techniques have been used to provide environmental protection and electrical insulation for hydrophones or even junction boxes.

2.8.2 Power Distribution Systems

For this and following discussions in this section, the physical reference will be for those electrical components located outboard of the personnel capsule. Electrical distribution and junction boxes are generally fluid-filled and pressure-compensated with a silicone or transformer dielectric fluid. Thin-walled, box-type construction techniques permit a considerable latitude of enclosure designs and shapes. The design rationale is that sea-water intrusion should be allowed to settle to an enclosure sump where it may be drained off. Hence, the fluid selected must minimize the dispersion or emulsification of sea water and/or must show a minimal degradation in electrical properties. Sea-water-leakage detectors should be installed as mentioned for hydraulic system enclosures, section 2.7.2.

Since no moving shaft penetrations are involved, these enclosures are usually unpressurized (zero Δp). As being very similar to hydraulic system compensation, a diaphragm or flexible membrane is installed over the enclosure opening or else the cover is gasket-sealed and

an external bag or bladder compensator is used. Since heat generation is not significant, PVC, fiber glass, or other plastic enclosure materials have been used.

To avoid failures at the electrical connector interface, many fluid-filled distribution or junction boxes employ cable-stuffing tubes. If the enclosure fluid is communicated throughout the cable internals, this is referred to as "self-hosing" cabling; both the electrical enclosure and associated cabling are depth/pressure-compensated. Cables may also be internally end sealed at the stuffing tube junction. Otherwise, connectors may be used to provide a natural interface between electrical enclosures and their required cabling. This simplifies the disassembly of electrical systems and components for maintenance, repair, or replacement.

To minimize problems of galvanic corrosion with electrical enclosures, grounding straps to the personnel capsule and sacrificial zinc anodes should be used. Zinc blocks may be used to provide sacrificial electrolysis for the entire vehicle (see specification MIL-A-180001).

Electrical distribution boxes contain circuit breakers or off-on contactors and motor controllers. Under "good" operating conditions, gas generation may not be a problem, but relief valve protection should be provided. Where gassing is heavy, a considerable amount of fluid may be relieved with the gaseous products. The volume previously occupied by escaped fluid will then be filled with gas. In the postdive checkout, this escaped fluid must be replaced.

It has been observed with at least four fluids - 1-cs silicone, MIL-H-56063, MIL-H-6083C, and Hoover(TM) Submersible 2 - that under pressures up to 13,500 psi, large volumes of gaseous products are dissolved in the fluid. However, when the pressure is released to 0 psig, these gases come out of solution very slowly. For the sake of illustration, perhaps 30% of the dissolved gases will be released within 1 hour. Within 4 hours, 70%; and after 16 hours, 99% of the gas may be released. Although it may not be of particular disadvantage, or a problem, personnel should anticipate this phenomenon when maintaining compensating systems. No open flames or smoking should be permitted in the area.

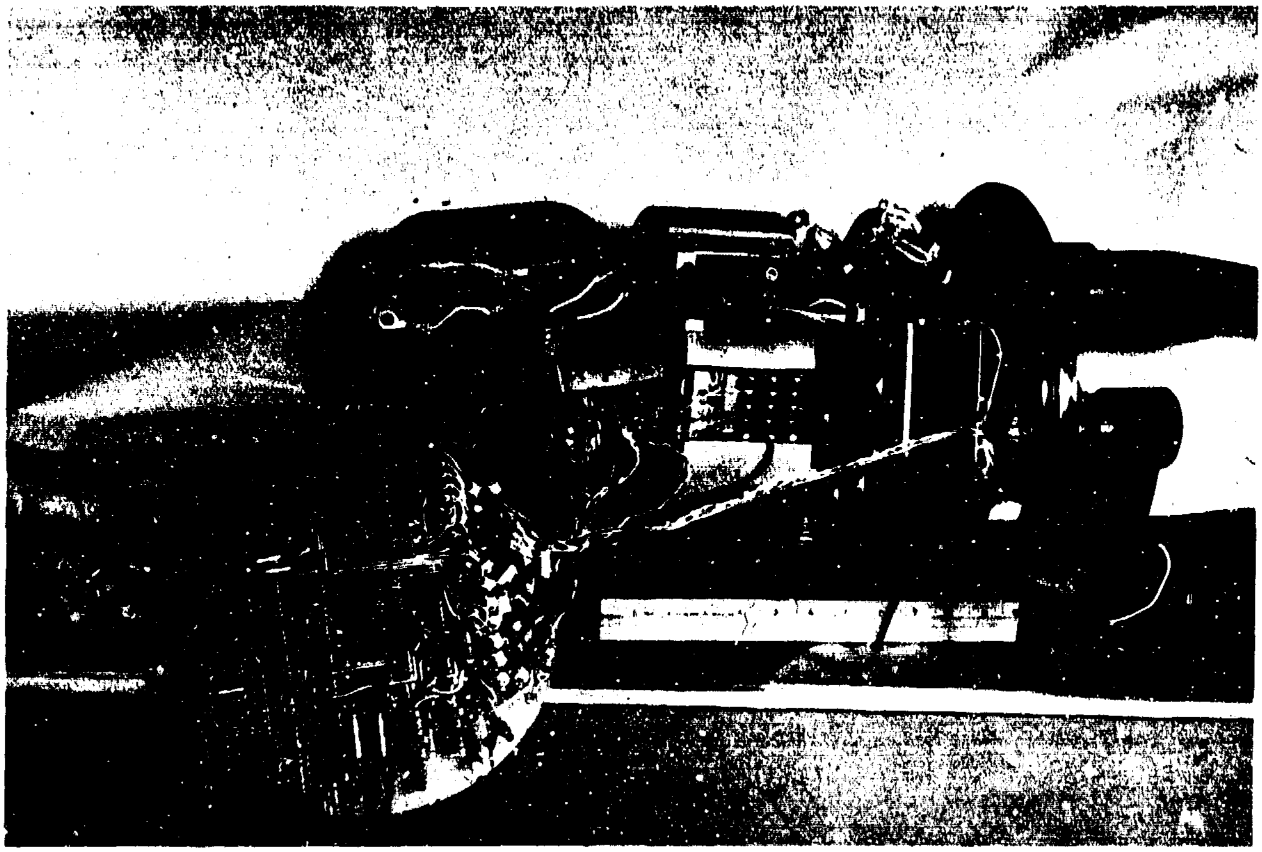
It has been reported that most fluids will break down badly and produce solid "clinker" products when exposed to the arcing of high current switches and circuit breakers. Silicone fluids have been more frequently used for electrical switching to reduce clinker formation. However, poor reliability and premature failures are still

observed in systems using silicone. The buildup of carbonaceous products and the subsequent degradation in fluid electrical insulation resistance is a serious problem since short-circuiting and grounding will occur in distribution boxes. Enclosure designs which provide considerable fluid volume will aid in heat transfer and the dilution and settling of dielectric breakdown products. In reference 30, extensive data are presented in a DOT handbook for the design and/or selection of mechanical circuit-interrupting devices for use in various fluids at pressures from atmospheric to 13,500 psi.

2.8.3 Electronic Enclosures

To provide a compact packaging arrangement and to make use of available pipe materials, cylindrical electronic enclosures are commonly used. End caps facilitate "windows" for components such as hydrophones, lights, and transducers and permit the mounting of electrical connectors at the other end of the cylindrical enclosure. A short length of cable may also be directly molded to the O-ring sealed end cap.

Since control stability and reliability have been major problems in the technology of fluid-filled, pressure-compensated ("wet") electronics, cylindrical configurations are frequently designed as thick-walled, "hard shell" structures. Presently, it appears desirable to design solid-state motor controllers so that the more oil- and pressure-sensitive electronic components and circuitry are hard-shelled. Electromechanical circuit interrupting devices, chokes, and other such large and readily fluid-immersed components may be housed in an adjoining fluid-filled section (see NAVSHIPS Deep Ocean Technology Status Report, Submersible Electric Drive Systems Development, June 1971). The use of an integral pressure-compensating device will facilitate a rugged, self-contained electronic package. Fluid-immersed, solid-state switching devices also show much promise in this type of application. Information on this investigation, "wet" electronics, and also on the development of encapsulated and submersible fuses will be published in follow-on chapters for reference 30.



Wet Electronic Motor Controller, Disassembled from Cylindrical Housing. (Arrow Indicates Circumferential Elastomeric Tube Compensator) (See Section 2.2.3)

CHAPTER III

FLUID CONSIDERATIONS AND COMPENSATOR DESIGN SCHEME

3.1 COMPENSATING FLUID CONSIDERATIONS

A fluid pressure-compensating system must compensate for changing physical conditions, both ambient and internal. The design scheme used in sizing a compensator is to estimate the most severe volume changes which will occur for any forseeable condition. Two physical characteristics must be known for a given compensating fluid: compressibility over the working pressure range and the coefficient of thermal expansion. A compensating fluid should not suffer excessive volume "shrinkage" as a function of increasing pressure. The term "bulk modulus" is the reciprocal of compressibility. Thus, a high bulk modulus (minimal compressibility) is desirable. Fluid specific gravity or weight may add appreciably to system weight. In consideration of functional performance and vehicle buoyancy, the sg of candidate compensating fluids is normally less than 1.0.

For most organic liquids, the higher the viscosity at atmospheric pressure, the higher the pressure dependence on viscosity. However, low viscosity silicone fluids have exhibited the smallest viscosity change with pressure while two polymer-containing fluids, MIL-H-5606B and MIL-H-6083C, had considerably less dependence on pressure than other oils of the same viscosity at atmospheric pressure. Viscosity and lubricity (lubricating ability) are very important fluid properties for machinery systems which must operate over a range of atmospheric pressure to ocean depths of 20,000 feet, where the environmental static pressure will reach 9,000 psig. With a safety factor of 1.5, a compensating fluid will see a maximum operational test pressure of 13,500 psig. The effect of these conditions, which increase fluid viscosity, imposes arduous restrictions on the choice of a fluid. A compensating system must readily respond, even when the viscosity has increased by a factor of ten at depth.

Problems will likely arise unless basic viscosity considerations are weighed during the initial drive system design. Some of these are as follows:

- Since the viscosity of fluids varies with temperature, pressure, and in the case of non-Newtonian fluids, with the rate of shear, a compensating fluid must be chosen which is best suited over the expected operating ranges.

- The viscosity must be high enough to provide adequate lubrication and low seal leakage, but low enough to prevent overheating and excessive viscous drag losses in hydraulic lines, motor "windage," gear pumping, etc.

- Bearing and gear loads generally must be reduced, as fluid viscosity is reduced, making the drive machinery larger for a given application. At the same time that viscous drag losses are reduced by using a lower viscosity fluid, power losses will increase as the size of the rotating machinery is increased. Thus, there is often an optimum choice of fluid viscosity to minimize system size and provide efficiency and reliability at depth.

- A fluid with the best viscosity range, however, may not provide a satisfactory range of other properties, such as high bulk modulus, corrosion resistance, dielectric properties, materials compatibility, fire resistance, lubricity, cost, etc. A compromise selection may be required. For instance, some silicone fluids of low viscosity have good dielectric strength and heat-transfer properties, but poor corrosion protection, poor lubrication, and low bulk modulus. Petroleum-based fluids have superior lubricating properties to those of silicone and under flow conditions and/or agitation, a fluid such as MIL-H-5606B may provide better dielectric strength with 9% sea-water contamination than a 10-cs silicone fluid with 1% sea water.

3.2 FLUID PROPERTIES

Subject to individual mission requirements, the following general fluid properties are desirable for system performance and safety.

- Specific gravity less than 1.0, where positive buoyancy is required.

- Favorable viscosity characteristics over the range of environmental temperature and pressure; 5-15 cs at 100° F, 0 psi.

- Satisfactory lubricating ability.

- Low compressibility (less than 10%) or high bulk modulus.

- Compatible with system materials.

- Corrosion protection when contaminated with 5%-10% sea water, by volume.

* High thermal conductivity, good heat-transfer properties.

* Good electrical properties.

* Fire resistance, low volatility; high flash point, 300° F minimum.

* Compatible with conditions of use: low toxicity, easily handled, chemically and thermally stable.

* Low vapor pressure; low gas solubility characteristics.

* Low thermal coefficient of expansion.

* Low foaming tendencies.

* Commercially available in quantity at low cost.

The order of importance of these fluid properties will depend upon specific performance requirements for the application and the system design and maintenance philosophy. Perhaps no fluid will ever be developed which completely satisfies all of the ideal or desirable compensating fluid properties. The DOT "Handbook of Fluids and Lubricants for Deep Ocean Applications,"[23] will be helpful in selecting a compensating fluid for a specific application. Fluid properties are discussed, evaluated, and tabulated data are presented for candidate fluids.

3.3 EMULSIFYING VERSUS NONEMULSIFYING FLUIDS

Since the specific gravity of seawater is greater than that of most candidate deep submergence fluids, sea-water intrusion or leakage will tend to "settle-out" of a non-agitated fluid, but not necessarily to a lowest point in the equipment enclosure. In the process of settling, seawater may penetrate electrical components, bearings, and other sensitive hardware. Also, the action of rotating machinery or other strong agitation will thoroughly distribute sea-water intrusion to contaminate the entire enclosure fluid volume. When a machinery system comes to rest and cools, there is also a tendency for ambient seawater to penetrate fluid films at dynamic seals.

An emulsifying fluid has the ability to disperse and encapsulate sea-water leakage so that very small

droplets are maintained in a stable suspension, or emulsion. "The concept of emulsifying the seawater has the advantage of encapsulating the water droplets in an "organic skin" of polar molecules which prevents or severely limits direct contact with metal surfaces."[36] Although an emulsified fluid usually has a characteristic "milkiness," the physical clarity of a fluid can be misleading. It is possible for 0.2%-0.3% sea water to be carried in a fluid without any noticeable emulsification.

For applications where the fluid is being continuously agitated and/or circulated, a greater percentage of sea-water leakage will be emulsified by a fluid. However, an emulsion may break in a matter of seconds after the agitation is stopped, and water droplets will fall out of solution. Most candidate compensating fluids do not have good emulsifying properties, but will emulsify some percentage of sea water under strong agitation. The stability of the emulsion is therefore more significant than the amount of sea water which is emulsified.

For fluid-immersed electrical equipment, there is a danger of short circuiting or catastrophic failure if sea-water leakage into the fluid occurs. "As little as 0.1% contamination by seawater reduces the resistivity of some fluids below suggested limits." [23] A fluid which forms stable emulsions with seawater offers protection against large droplets or concentrations of seawater bridging an electrical gap. If sea-water contamination in an electric drive motor is emulsified into extremely small droplets, the motor may still be operable, even though dielectric heating and loss of efficiency may occur.

In the "Handbook of Fluids and Lubricants for Deep Ocean Applications,"[23] an emulsion stability method is described by which fluids are classified for electrical equipment service according to the time required for water separation. A tentative standard is as follows:[23]

Classification	Time Required for Water Separation, minutes
A. Suitable for use with motors	5 or more
B. Questionable for use with motors	1-5
C. Unsuitable for use with motors	<1
D. Suitable for contactors, switches, etc	No emulsion stability requirement

During a DSV mission, electric motors and associated machinery may be frequently started and stopped. Thus, it is desirable for a fluid to maintain a stable emulsion with sea-water intrusion in order to afford protection for electric motors during nonoperating periods. A sea-water fluid emulsion which will not break within 5 minutes is considered reasonably stable, and sea water is not likely to coalesce between conducting surfaces and cause electrical shorting.

A supposedly nonemulsifying fluid, MIL-H-5606B, will emulsify up to 1% sea water in a continuously agitated system. Any fluid with a sulfonate corrosion inhibitor will emulsify sea water to some extent. An emulsifying fluid, MIL-H-6083C, will emulsify 4%-5% sea water in a system which is repeatedly agitated and then stopped. For continuous circulation, as in a motor and speed reducer, this fluid will emulsify up to 10% sea water.

3.3.1 Fluid Selection

A broad decision should not be made for either alternative without giving each component of the system individual consideration. For obvious reasons, a systems approach is necessary in fluid selection from the design concept stage. The operational requirements, length of mission, and system maintenance are necessarily unique for each deep ocean application.

To enumerate some of the parameters, consider two fluids of nearly the same viscosity, with the nonemulsifying fluid having, in general, fewer additives. The nonemulsifying fluid will afford less corrosion protection, but will have better dielectric qualities than an emulsifying fluid. Fluids with emulsifier additives logically have hygroscopic properties which may even encourage sea-water intrusion. "The need for fluids and lubricants with corrosion-protection properties and improved lubricating ability has led to the formulation of products which contain polar additives and those in which water is soluble or with which water is miscible. In addition to lowering the resistivity and dielectric breakdown voltage, the polar materials also decrease the efficiency of an electric motor by transformation of electrical energy into heat energy in a nonsinusoidal alternating-current system."*

Polar additives, which tend to lower fluid electrical properties, also may lessen the lubricity or ability of a fluid to lubricate. The findings of a recent report[35] indicate that finely dispersed water in an emulsifying fluid, MIL-H-6083C, acted as an antiscuffing agent, being superior to MIL-H-5606B in this respect. However, 5606B demonstrated better overall wear resistance for steel on steel sliding, even in the presence of 1% sea water and humid oxygen. The calcium or barium sulfonate corrosion inhibitor, an emulsifying agent in 6083C which tends to continuously supply finely dispersed water droplets to the sliding contact zone, seems to increase the wear rate. However, the worst increase in wear rate occurs when water has entered an uninhibited, nonemulsifying fluid.

If 10% or greater seawater is present in an emulsifying fluid, the emulsion formed will tend to increase the apparent fluid viscosity. A further consideration is that hard oxide corrosion products in a fluid will accelerate bearing failure. A fluid with a nonemulsifying corrosion inhibitor additive is then desirable for certain applications.

In this respect, Tellus(TM) 11, 15, and 27 are good compromise fluids and have potential as "general-purpose" compensating fluids. The corrosion inhibitor additive is nonemulsifying. Fluid dielectric properties are good for Tellus(TM) 11 and 15. Corrosion-inhibiting

*Reference 23, page I-7.

properties are moderately good for both ferrous and nonferrous metals. These fluids also passed the 48-hour ASTM D-665 stirred rust test, whereas MIL-H-5606B failed.23

Materials compatibility with sea water and with the compensating fluid must be a high priority consideration, especially for electrical insulating materials. Grounding and low insulation resistance are critical problems in both d-c and a-c equipment. An insulation system which can withstand sea-water contamination will necessarily be more complex, bulkier, heavier, and more expensive. Thus, housing dimensions must be increased and more compensating fluid will be required. With bulkier insulation, heat transfer may also be a problem.

The "tradeoff" situation that arises is then not only whether to use an emulsifying or nonemulsifying fluid, but what fluid characteristics are most critical for the reliability of the system. A general degradation of a compensating fluid will take place if excessive heating occurs or if contamination occurs, but acceptable limits of seawater and solid contamination have not been established. Moreover, various contaminants, loading and wear of gears and bearings, and electrical arcing have additive effects on fluid electrical properties which are not known.

3.3.2 System Maintenance

The ramifications of sea-water intrusion must be weighed for each candidate design. The materials specifications will differ greatly as to the predicted fluid environment of the internal system components. A designer may "map out" a scheme of questions to be answered, such as:

- Will the compensating fluid adequately protect and lubricate the components?

- Will the electronics and insulation system withstand sea water?

- For this particular application, at what intervals will the system receive fluid maintenance? Will contamination be drained off and fluid replenished or will the system be drained and refilled?

Military Fleet submarines must be reliable for long missions and extended submergence. Outboard, fluid-filled systems may receive maintenance after months of submergence time and the fluids used must provide adequate corrosion inhibition. The military specification may call for electrical insulation systems to be sea-water floodable, as a worst operating condition. This is the most conservative, however, expensive, alternative. Another approach is to select a fluid which will emulsify sea water and enable the insulation system to "stand up" longer than with a nonemulsifying fluid. If the insulation materials are essentially incompatible with water, an emulsifying fluid should not be used.

A small, deep-diving submersible will have a different set of parameters for fluid maintenance. Mission time is normally a few hours. During the postdive checkout, enclosure drain ports may be opened to remove sea-water leakage and to replenish compensating fluid. A fluid reconditioning apparatus is usually employed to circulate, filter and heat the fluid so that sea water and particulate contaminants are removed (see section 4.7).

For this application, a nonemulsifying fluid appears to be desirable in that sea-water contamination will tend to separate from the fluid. The condition of the fluid may be readily inspected between dives. Systems may be drained and replenished if necessary. Due to the short submergence time, corrosion protection for metals and moving parts is not as critical, especially where the fluid is agitated and circulated during the majority of the mission time. From this standpoint, fluids which are nonemulsifying, have good electrical properties, and yet provide some corrosion protection (as the previously mentioned Tellus™ fluids) deserve particular consideration. To repeat, a very critical consideration for electrical equipment is whether the insulation system is better in an emulsifying fluid, considering some seawater leakage, or is it more advisable to drain off sea-water leakage between dives?

A maintenance procedure which has been used for hydraulic systems of Fleet military submarines is to use a good detergent oil as a flushing agent. MIL-H-6083C is a good flushing oil for use with MIL-H-5606B. Not only will 6083C form a stable emulsion with sea water, but it is quite compatible with 5606B so that insoluble residues will not be left in the hydraulic system. This approach would also be useful for flushing DSV fluid power systems and various equipments, such as speed reducers.

To date, fluid reconditioning or reclaiming processes have been much less effective with fluids which have formed emulsions with sea water in actual use conditions than with fluids which have not formed emulsions. Even with a nonemulsifying fluid, a reconditioning process may not restore original fluid properties to a full extent (see sections 4.7.4 and 4.7.5).

Housings for electrical distribution systems and power conversion and control components are usually compensated with a nonemulsifying fluid. Low-viscosity/low-compressibility fluids should be selected which have good electrical properties. The fluid receives little agitation in this application and the design philosophy calls for essentially no sea-water contamination.

3.4 TEMPERATURE AND PRESSURE ENVIRONMENT

The following temperature ranges represent estimated environmental extremes for deep ocean systems, from the tropics to arctic latitudes:

- Ambient Air (storage or transportation conditions) - -40° to 140° F.
- Ambient Sea water - 28° to 90° F.
- Drive Machinery - Environmental temperature to 190° F.

The depth distribution of the world's oceans is such that a drive system designed for missions at 20,000-foot depths will provide the capability for submersibles to cover over 98% of the ocean floor. The general trend is for sea-water temperature to decrease as the depth increases. Near-surface Antarctic waters are perhaps the coldest, where temperatures of -1.89° C (28.58° F) have been measured. "The North Atlantic is the overall warmest of the ocean areas and the South Atlantic is the coldest. The North Pacific has the highest average surface temperature (19.68° C or 67.42° F) while the South Atlantic has the lowest (11.91° C or 53.44° F). The average bottom temperatures vary from 0.74° to 1.56° C (33.33° to 34.81° F) at 5000 meters."[39] An overall average temperature for all oceans is about 38° F. For each location in an ocean, mass properties of sea water are unique. Temperature, salinity, and density vary throughout the vertical water column. To calculate ambient pressures for depths up to 20,000 feet, 0.45 psi/ft is a

reasonably accurate, conservative factor. The table presented below is based on average temperature-depth data for all latitudes of the Atlantic, Pacific, and Indian Oceans.[39]

Table of Average Ambient Sea-Water Temperatures Versus Depth and Pressure for the Open Ocean

Avg Ocean Temp, ° F	Depth ft	Depth meters	Pressure psig (approx)
34	35,800	10,915	16,400
34	30,000	9,146	13,700
34.5	20,000	6,098	9,000
34.5	15,000	4,573	6,750
35	12,000	3,659	5,400
35.5	10,000	3,049	4,500
35.7	8,000	2,439	3,600
37	6,000	1,829	2,700
37.5	5,000	1,525	2,250
38	4,000	1,220	1,800
40	3,000	915	1,350
43	2,000	610	900
46	1,000	305	450
53	500	153	225
61	surface	0	0

For DSV systems, the test depth is usually 1.5 x (operational depth). A 12,000-foot DSV will have a maximum operational test requirement of 18,000 feet and severe constraints are imposed on all systems. Fluid compressibility, viscosity, and density increase significantly. This establishes one set of design extremes for determining compensator volume allowances.

3.5 COMPENSATOR DESIGN SCHEME

The requirements for a direct-drive, DSV thruster motor are used as an example to establish a general procedure for sizing and designing compensators. The design calls for a rugged, pod-mounted drive assembly to turn a 19-inch diameter, 13.5 inch pitch propeller at 600 rpm. A compensator must be sized as a function of extremes in physical conditions and also to provide for fluid leakage and sea-water intrusion via the shaft seal. An internal, spring-biased, rolling diaphragm compensator will be designed so that the drive motor and compensator housing are integral for pod mounting.

Motor Requirements

Motor, thruster, ac (NA)*

Rated 8 hp at 600 rpm, direct drive (NA)

Maximum thrust: 370 pounds (NA)

Maximum Torque: 70 ft-lb (NA)

Operational depth: 10,000 feet, 4,500 psig

Overall pressure range: vacuum (fill) to 6,750 psig

Overall temperature range: -40° to 190° F

Estimated weight: 290 pounds, in air (NA)

Motor fluid fill volume, less compensator: 300 in.^{3}

Test depth: 15,000 feet, 6,750 psig

3.5.1 Design Parameters. (Also, it may be helpful to refer to compensating system survey outline as used for selected surveys in section 5.3)

- Compensating Fluid Selected

  Type: MIL-H-5606B, petroleum base, red, hydraulic

*(NA) - Not applicable or required

Coefficient of Thermal Expansion: 0.00041/° F

Viscosity: 13.80 cs at 100° F, 0 psig.[41] Laminar flow conditions in motor case, thus figure is for low shear rate, highest kinematic viscosity

Compressibility Factor: $V/V_0 = 0.9765$ at 6750 psig, 35° F.[41]

- Compensator Material

  Case: 316 L stainless steel (welded fabrication)

  Spring: Exposed to sea water, will provide 1-2 psi positive bias. Spring material type 17-7PH stainless steel

  Elastomer: Rolling diaphragm of dacron fabric and Buna-N or Nitrile. Good compatibility with MIL-H-5606B

- DSV Storage, Transportation Conditions

  Pressure: Atmospheric 14.7 psi or less (air transportable)

  Temperature Extremes:

-40° F (on deck of support ship in Arctic)	to	140° F (metal warehouse in tropics)

- Operational Characteristics

  Surface conditions

  Maximum operating temperature - 190° F

  Maximum ambient sea-water temperature - 90° F

  Depth conditions

  Estimated maximum operating temperature - 150° F (overload)

Lowest ambient sea-water temperature
34.5° F

- Shaft Seal Leakage (estimated maximum) - 30 cc/hr. for mechanical end face seal. Although leakage allowance will hinge on mission requirements, for mechanical contact seals, a 5% volume allowance should be more than adequate in most applications. Lip seal leakage is often greater than that of mechanical contact seals (also see reference 34).

Volume change for 8-hour mission:

$$30\text{cc/hr} \times 8\text{ hr} = 240\text{cc or } 14.64\text{ in.}^3$$

$$\frac{14.64\text{ cu. in.}}{300\text{ cu. in.}} = 0.0488\text{ or } 4.88\%\text{ fluid}$$

loss

- Sea-Water Intrusion (estimated)- 4 cc/hr

Volume change for 8-hour mission:

$$4\text{ cc/hr} \times 8\text{ hr} = 32\text{ cc, } 2\text{ in.}^3$$

$$\frac{2\text{ in.}^3}{300\text{ in.}^3} = 0.00668\text{ or } 0.668\%\text{ contamination}$$

gained

3.5.2 Design Scheme

Assume that the motor is nonoperating at the test depth of 15000 feet. Average ambient conditions at this depth are 6750 psig and 34.5° F (refer to the table in section 3.4). All fluid volume calculations will be referenced to these maximum depth conditions, with the rolling diaphragm compensator at a minimum stroke and with MIL-H-5606B fluid completely filling the 300 in.^{3} of the motor. The most severe change in physical conditions, and thus compensator volume, will occur between condition 1, at depth, and condition 2, on the ocean surface, where the motor is operating at the maximum foreseeable temperature and low ambient pressure. Fluid compressibility data for MIL-H-5606B at 35° F is available in reference 41.

	Condition 1 Maximum Depth	Condition 2 Surface
Ambient Temperature	ta_1 = 34.5° F	ta_2 = 90° F
Fluid Temperature	t_1 = 34.5° F	t_2 = 190° F
Ambient Pressure	P_1 = 6750 psig	P_2 = 0 psig
Fluid Volume	V_1 = 300 in.^{3}	V_2 = ?

The compressibility of a compensating fluid can be thought of as a built-in "spring effect" so that as the pressure decreases, fluid volume will increase. At the ocean surface, due to compressibility alone:

$$V_2 = V_1/0.9765$$

For 5606B, the coefficient of thermal expansion B is 0.00041/° F. Considering both temperature and pressure effects on volume, from condition 1 to 2:

$$V_2 = V_1/0.9765 + B(t_2 - T_1)\ V_1$$

$$V_2 = 300/0.9765 + 0.00041\ (190 - 34.5)\ 300$$

$$V_2 = 326.4 \text{ in.}^3$$

$$\text{Volume Change}_{1-2} = \frac{326.4 - 300}{300} = 0.0880 = 8.80\%$$

$$V_{1-2} = 26.4 \text{ in.}^3$$

Volume allowances must also be made for storage and transportation conditions. The lowest estimated temperature requirement is for an Arctic, in-air temperature of -40° F. This calculation will be referenced to condition 1, at minimum compensator stroke.

	Condition 1 Maximum Depth	Condition 3 Arctic Storage
Ambient, Fluid Temperature	$t_1 = 34.5°$ F	$t_3 = -40°$ F
Ambient Pressure	$p_1 = 6750$ psig	$p_3 = 0$ psig (sea level, atmos)
Fluid Volume	$V_1 = 300$ in.	$V_3 = ?$

$$V_3 = V_1/0.9765 + 0.00041(t_3 - t_1)\ V_1$$

$$V_3 = 300/0.9765 + 0.00041(-40 - 34.5)\ 300$$

$$V_3 = 307.22 - 9.16$$

$$V_3 = 298.06 \text{ in.}^3$$

$$\text{Volume change}_{1-3} = \frac{298.06 - 300}{300} = -0.00646 = -0.65\%$$

$$V_{1-3} = -1.94 \text{ in.}^3$$

Thus, storage or transportation conditions in the Arctic would more than deplete the compensator at minimum stroke (bottomed out). The volume at condition 3 would be almost 2 in.^{3} less than the minimum volume at condition 1 (see later discussion of volume allowances).

The 190° F maximum operating fluid temperature for condition 2 exceeds the 140° F tropic storage requirement; therefore, this volume change has already been covered.

From the design parameters, some leakage at the rotary shaft seal is anticipated for an 8-hour mission of the DSV. Sea-water intrusion may amount to 2 in.^{3} while fluid leakage into the sea water is estimated to be 14.64 in.^{3}

At minimum compensator stroke, a rolling diaphragm configuration will have a small inactive fluid volume in the "roll" of the elastomeric diaphragm, etc. It is estimated that this dead band allowance or inactive fluid volume will amount to 7 $in.^3$

3.5.3 Volume Allowance Rationale

Since compensators have often been undersized rather than oversized, a safety factor of 1.4 will be used to insure an adequate range of fluid volume. To sum the volume change calculation for various conditions,

$$V_T = V_{1-2} + |V_{1-3}|^* + V_{leakage} - V_{sea\text{-}water\ intrusion} + V_{dead\ band}$$

$$V_T = 26.4 + 1.94 + 14.64 - 2 + 7$$

$$V_T = 47.98 \text{ or } 48 \text{ in.}^3$$

$$V_{comp} = \text{safety factor} \times V_T$$

$$V_{comp} = 1.4\ (48) = 67.2 \text{ in.}^3$$

Note that this scheme of calculations is not complete in that the volume change of the fluid in the compensator must also be considered. Using the same scheme of calculations, temperature and pressure effects on V_T = 48 in. amount to about 5 $in.^3$ Since V_T was multiplied by a safety factor of 1.4, this will easily accommodate an additional 5 $in.^3$ volume change. The use of a safety factor generally affords protection for: trapped air or voids in the system, entrained air in the compensating fluid, loss of fluid through the relief valve during gassing, excessive fluid leakage at the rotary shaft seal, and unforseen extreme operating conditions.

If the compensator spring provides a minimal 1-2 psid positive bias, this will allow the longest contamination-free operating time following a condition which initiated excessive leakage. Should the compensator volume be completely depleted, check valve 1 will

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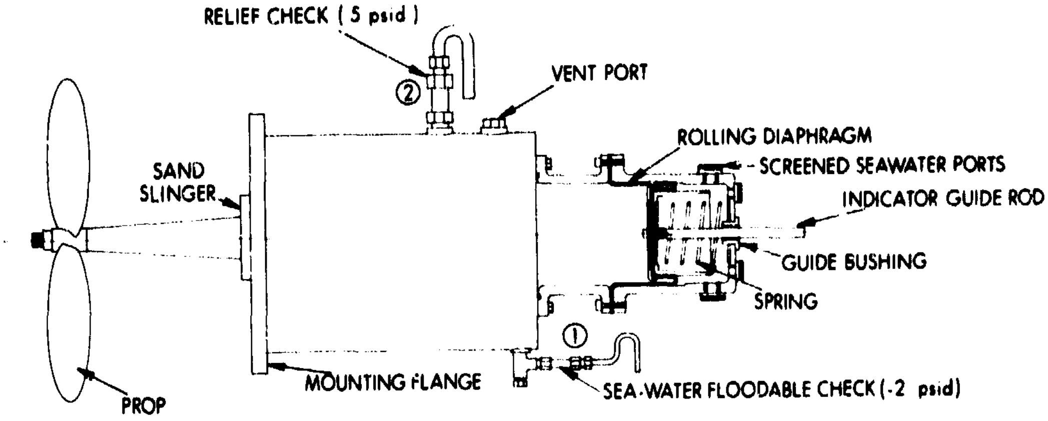
* V_{1-3} is considered as an absolute value since this volume change at -40° F would deplete the compensator.

permit seawater to directly compensate the a-c motor as a "fail-sick" worst condition. When the internal case pressure becomes -2 psid or 4.08 in. Hg vacuum, check valve 1 will let in sea water. The No. 2 relief check pressure is set at 5 psid so that gas and/or fluid will be relieved, rather than to "blow out" the compensator elastomer or damage the shaft seal.



Thruster Motor with Integral Rolling Diaphragm Compensator

It appears that the best approach is to select a rolling diaphragm such that, at 2/3 stroke, the compensator volume will be about 67.2 $in.^3$, the calculated V_{comp}. This will permit an adequate range of "travel" and enhance compensator life.

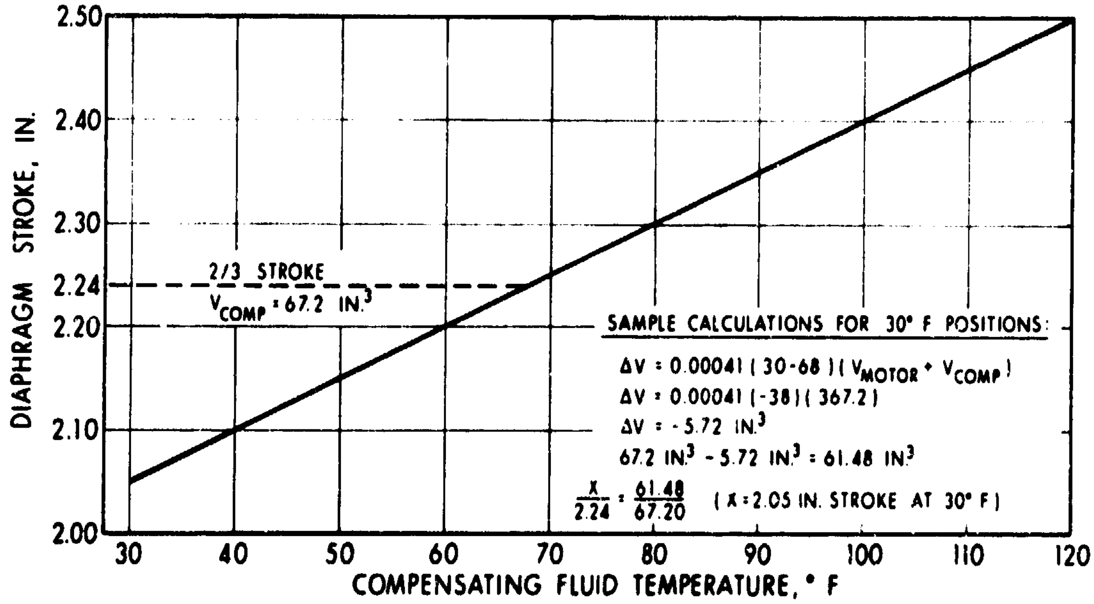
An available rolling diaphragm has an effective area, A_e = 30 $in.^2$ Let the full diaphragm stroke be L. For a 2/3-stroke position with a compensator volume of 67.2 $in.^2$

$$2/3\ L = \frac{V_{comp}}{A_e}$$

$$2/3\ L = \frac{75\ in.^3}{30\ in.^2} = 2.24\ in.$$

$$L = 3.36\ in.$$

To establish a reference temperature condition for fluid filling, the motor will be filled so that the compensator is in mid-stroke position at atmospheric room conditions (68° F). The compensating fluid will undergo pressure and temperature cycling, and it is necessary that the compensator position be adjusted for each fluid filling temperature as per the example in the accompanying graph. The position of the compensator diaphragm may be visually monitored by the projection of the indicator-guide rod. At nearly minimum stroke, the 17-7 PH stainless steel spring will provide a positive bias of about 1 psid. If enough fluid expansion or gassing occurs that the rolling diaphragm is completely distended (spring-compressed), the overpressure at this maximum stroke will be 2 psid.



Corrected Compensator Position for Fluid Fill
Temperature, Sea Level Atmosphere
Reference Temperature 68° F

The sea-water-flooded spring cavity should have a minimum of two baffled and/or screened ports, as shown. Grit and marine life must be filtered from the surrounding seawater and yet the ports should not be susceptible to clogging. Thus, an innovative filtering scheme is necessary to protect the elastomeric diaphragm and internal cavity.

3.6 SELECTED FLUID DATA

Chemical and physical properties of compensating fluids are affected in varying degrees by the low ambient temperature and the high ambient pressure of the ocean at operating depth. Certain excerpts of fluid properties data are included as a design aid, along with various rankings of fluids and thermal expansion coefficients. For more thorough fluid data, references 3, 23, 24, and 41 are recommended.

3.6.1 Summary List of Fluids and Lubricants Tabulated*

Specification or Trade Name	Other Designation	Base Fluid Composition	Application: Power Transmission	Lubrication	Motor Immersion	Switching Component Immersion	Nonmoving Electrical Equipment Immersion
Federal Specification Products							
VV-I-530a	Transformer Oil	Petroleum	-	-	Q	KP	KP
VV-D-001078(10 cs)	Damping Fluid	Silicone	Q	Q	KQ	KQ	KP
VV-D-001078(50 cs)	Damping Fluid	Silicone	KQ	Q	Q	Q	Q
Military Specification Products							
MIL-H-5606B	Aircraft Hydraulic Fluid	Petroleum	KP	KP	KP	P	P
MIL-J-5624F	JP-5	Petroleum	-	KQ	KQ	Q	Q
MIL-L-6081C, Grade 1010	Jet Engine Lubricating Oil	Petroleum	KQ	KQ	KQ	KQ	Q
MIL-H-6083C	Aircraft Hydraulic System Preservative	Petroleum	K	KQ	KQ	KQ	KQ
MIL-L-6085A	Aircraft Instrument Oil	Synthetic	KQ	KQ	KQ	Q	Q
MIL-L-7808G	Gas Turbine Lubricating Oil	Synthetic	-	KQ	Q	Q	Q
MIL-L-7870A	-	Petroleum	-	K	Q	Q	Q
MIL-C-8188C	Gas Turbine Engine Preservative	Synthetic	KQ	KQ	Q	Q	Q
MIL-F-17111	Ordnance Hydraulic Fluid	Petroleum	Q	P	-	-	P
MIL-L-17672, MS 2110-TH	Turbine Oil and Hydraulic Fluid	Petroleum	KQ	KQ	Q	Q	P
MIL-S-21568A	Damping Fluid	Silicone	Q	Q	KQ	KP	KP
MIL-L-23699A	Aircraft Turboprop and Turboshaft Lubricant	Synthetic	-	KQ	-	-	-
MIL-H-27601A	Aircraft High Temperature Hydraulic Fluid	Petroleum	-	-	-	-	-
MIL-H-46004	Missile Hydraulic Fluid	Petroleum	KQ	-	-	-	-
MIL-H-81019B	Aircraft and Missile Hydraulic Fluid	Petroleum	P	Q	-	-	P
Proprietary Fluids							
Fluid Code A	Sea-water Emulsifying Fluid, Type I	Petroleum	KQ	KQ	Q	Q	Q
Fluid Code B	-	Petroleum	KP	KQ	Q	Q	Q
Fluid Code C	Proposed Specification MIL-H-25598 Missile Hydraulic Fluid	Petroleum	KP	KQ	Q	Q	Q
Fluid Code D	Traction Drive Fluid	Petroleum	-	-	-	-	-
Fluid Code E	-	Petroleum	-	KQ	KQ	-	P
Fluid Code F	-	Petroleum	P	P	-	-	P
Fluid Code G	-	Petroleum	P	P	-	-	P
Fluid Code H	-	Petroleum	P	P	-	-	P
Fluid Code J	USP Mineral Oil	Petroleum	-	Q	KQ	KQ	KP
Fluid Code K	NF Mineral Oil	Petroleum	-	Q	-	-	-
Fluid Code L	Lubricity Improved Silicone	Silicone	Q	Q	KQ	KP	KP
Fluid Code M	-	Petroleum	-	P	Q	Q	Q
Fluid Code N	Sea-water Compatible Water Glycol	Water	-	Q	Q	Q	Q

P - Possible use
K - Known or attempted use
Q - Questionable for use in this application
- (blank) - Insufficient information available for assessment of use

*McQuaid and Brown,[23] page III-4.

3.6.2 Measured Kinematic Viscosity, centistokes*

Fluid	Temperature °F	Pressure, psig 0	3,000	5,000	8,000	10,000	15,000	20,000
VV-I-530a	35	57.00	99.99	154.93	259.90	370.05	912.00	1,015.71
	100	9.65	16.78	20.47	30.57	39.21	76.67	141.26
	150	4.86	6.44	8.03	11.34	13.35	21.62	36.21
MIL-L-6081C	35	53.73	93.56	139.66	228.52	323.58	778.59	1,834.46
	100	10.08	15.43	19.77	28.98	37.37	69.84	129.94
	150	5.85	6.50	7.81	11.15	13.35	21.98	36.23
MIL-L-17672B	35	338.14	628.44	984.47	1,776.20	2,626.17	6,800.[1]	17,400.[1]
	100	35.68	59.23	79.88	125.45	167.04	334.05	660.82
	150	12.69	18.71	24.42	35.15	44.76	80.04	140.93
MIL-H-5606B high shear	35	29.85	43.44	52.89	76.16	92.62	152.82	264.44
	100	12.26	16.31	19.21	24.90	29.32	42.90	62.55
	150	7.21	9.74	11.39	14.24	16.07	22.88	31.65
MIL-H-5606B low shear	35	40.50	60.55	86.66	113.48	147.11	270.53	504.82
	100	13.80	18.39	21.87	32.33	38.78	58.41	92.59
	150	8.60	11.53	14.01	18.82	23.53	35.07	52.38
MIL-H-6083C high shear	35	30.25	43.35	55.28	77.65	101.64	173.86	311.67
	100	11.78	15.66	18.64	24.30	28.59	42.57	63.64
	150	7.27	9.24	10.75	13.45	14.02	21.99	30.30
MIL-H-6083C low shear	35	49.00	74.17	92.71	136.79	175.31	325.62	591.15
	100	15.80	21.42	26.05	33.94	39.51	61.13	93.77
	150	9.40	11.92	14.62	18.92	22.13	33.53	48.27
MIL-S-21568A	35	1.19	2.19	2.95	4.29	5.11	8.12	11.56
	100	0.76	0.96	1.25	1.73	1.97	2.91	3.92
	150	0.44	0.68	0.83	1.19	1.31	1.77	2.46
VV-D-001078	35	17.86	25.67	31.98	44.45	49.44	78.64	123.52
	100	8.66	12.37	14.99	20.16	23.82	35.52	51.12
	150	4.75	8.30	10.11	12.63	15.00	21.90	30.55
MIL-H-5606B base fluid	35	8.38	12.68	16.03	23.81	30.88	56.96	106.20
	100	2.81	4.04	5.11	7.10	8.35	12.86	20.40
	150	1.39	1.95	2.37	3.19	3.99	5.95	8.89
MIL-H-6083C base fluid	35	9.84	14.85	18.36	27.51	35.28	65.62	119.25
	100	3.53	4.89	5.95	7.76	9.03	13.99	21.48
	150	2.32	2.87	3.52	4.56	5.36	8.09	11.65

[1] Extrapolated values.

*Ventriglio, *et al*,[41] page 6.

3.6.3 Summary of Findings*

For comparison purposes, the isothermal compressibility, (percent volume reduction) data are summarized in the tabulation below. The data permit the fluids to be assigned to two, or possibly three, levels of compressibility. As expected, petroleum fluids have lower compressibility than silicone fluids. The VV-I-530a, MIL-L-6081C, and MIL-L-17672B petroleum fluids (containing only small amounts of additive agents, if any) have the least change in volume with change in pressure. The MIL-H-5606B and MIL-H-6083C petroleum-based fluids (each containing about 9% by weight of polymethacrylate-type viscosity-improving polymer and smaller amounts of other additive agents) have slightly higher volume changes. Polymeric materials would be expected to increase the compressibility over that of the base fluid, due to the less compact structure of the polymer molecule. However, the possibility still exists that the small compressibility increase found in the polymer-containing fluids (MIL-H-5606B and MIL-H-6083C)

Percent Volume Reduction Due to
Isothermal Compressibility

Temperature °F	Pressure psig	VV-I-530a Petroleum	MIL-L-6081C Petroleum	MIL-L-17672B Petroleum	MIL-H-5606B Petroleum	MIL-H-6083C Petroleum	MIL-S-21568A Silicone	VV-D-001078 Silicone
35	3,000	1.00	1.04	1.03	1.12	1.10	2.21	1.88
	5,000	1.61	1.64	1.64	1.81	1.82	3.55	2.97
	8,000	2.42	2.54	2.48	2.76	2.79	5.19	4.40
	10,000	2.92	3.08	3.00	3.33	3.39	5.61	5.21
	15,000	4.03	4.22	4.11	4.60	4.65	8.20	6.98
	20,000	5.03	5.29	5.15	5.77	5.81	9.85	8.49
90	3,000	1.12	1.18	1.14	1.29	1.29	2.75	2.11
	5,000	1.84	1.91	1.80	2.07	2.08	4.21	3.38
	8,000	2.73	2.91	2.73	3.12	3.17	6.11	4.95
	10,000	3.29	3.48	3.29	3.75	3.79	7.17	5.82
	15,000	4.50	4.73	4.53	5.12	5.15	9.21	7.66
	20,000	5.53	5.86	5.62	6.32	6.33	10.93	9.22
180	3,000	1.45	1.42	1.40	1.60	1.62	3.65	2.62
	5,000	2.30	2.29	2.17	2.52	2.58	5.38	4.02
	8,000	3.41	3.42	3.19	3.66	3.74	7.56	5.72
	10,000	4.06	4.10	3.83	4.40	4.93	8.72	6.75
	15,000	5.38	5.53	5.15	5.94	5.93	11.00	8.74
	20,000	6.50	6.80	6.34	7.21	7.24	13.00	10.51

*Ventriglio, et al,[41] page C-2.

may be due to differences in the petroleum oil bases rather than to the polymers. Silicone fluids have significantly greater changes in volume with pressure, amounting to almost twice that shown by the petroleum fluids.

3.5.4 Calculation of Densities from Compressibility Data*

Fluid densities for use in viscosity calculations were derived from the isothermal compressibility data delivered by the San Francisco Bay Naval Shipyard. The compressibility data are expressed as percent reduction of the initial volume, that is, at a given temperature.

$$-\Delta V = \left[\frac{V_o - V}{V_o}\right] 100$$

where, for a given temperature,

$-\Delta V$ = percent volume reduction or the isothermal compressibility

V_o = specific volume of liquid at atmospheric pressure (ml per gram)

V = specific volume of liquid at P psig (ml per gram)

Specific volumes at 32° F were not provided in the data mentioned above. They were obtained from atmospheric pressure measured density data of these fluids produced by Esso Research and Engineering Company under contract with this activity. V may then be calculated from the above equation, and density, ρ, is the reciprocal of specific volume, that is,

$$\rho = \frac{1}{V}.$$

*Ventriglio, et al,[41] page C-3.

Densities in grams per milliliter are given below

Densities, grams per milliliter

Temperature °F	Pressure kpsi	Fluids						
		VV-I-530a	MIL-L-6061C	MIL-L-17672B	MIL-H-5606B	MIL-H-6083C	MIL-S-21568a	VV-D-001078
35	0	0.8947	0.8812	0.8866	0.8659	0.8698	0.8383	0.9572
	3	0.9037	0.8905	0.8958	0.8756	0.8795	0.8543	0.9788
	5	0.9093	0.8959	0.9014	0.8818	0.8859	0.8639	0.9924
	8	0.9169	0.9041	0.9092	0.8908	0.8948	0.8769	1.0095
	10	0.9216	0.9092	0.9140	0.8957	0.9003	0.8844	1.0062
	15	0.9323	0.9200	0.9213	0.9076	0.9122	0.9012	1.0892
	20	0.9421	0.9304	0.9347	0.9189	0.9234	0.9161	1.1093
100	0	0.8700	0.8567	0.8632	0.8401	0.8445	0.8009	0.9238
	3	0.8804	0.8671	0.8735	0.8516	0.8560	0.8189	0.9514
	5	0.8868	0.8740	0.8796	0.8585	0.8630	0.8299	0.9663
	8	0.8952	0.8831	0.8879	0.8679	0.8729	0.8435	0.9859
	10	0.9006	0.8884	0.8933	0.8739	0.8786	0.8516	0.9976
	15	0.9122	0.9003	0.9050	0.8867	0.8915	0.8687	1.0206
	20	0.9174	0.9112	0.9157	0.8981	0.9029	0.8840	1.0404
150	0	0.8511	0.8380	0.8468	0.8202	0.8283	0.7709	0.8973
	3	0.8636	0.8501	0.8588	0.8335	0.8388	0.7916	0.9313
	5	0.8711	0.8576	0.8656	0.8414	0.8472	0.8032	0.9484
	8	0.8812	0.8677	0.8747	0.8513	0.8573	0.8177	0.9708
	10	0.8871	0.8738	0.8805	0.8580	0.8680	0.8267	0.9831
	15	0.8995	0.8871	0.8928	0.8720	0.8773	0.8447	1.0092
	20	0.9103	0.8992	0.9041	0.8839	0.8898	0.8614	1.0314

kpsi - thousand pounds per square inch.

3.6.5 Ranking of Seven Fluids[41] with Respect to Fluid Properties

Kinematic Viscosity, cs	Isothermal Compressibility	Density 1 Atm, 100° F	
MIL-S-21568A (1 cs) (low)	MIL-L-17672B	MIL-S-21568A (1 cs)	0.8009
VV-D-001078 (10 cs)	VV-I-530a	MIL-H-5606B	0.8401
MIL-H-5606B	MIL-L-6081C	MIL-H-6083C	0.8445
MIL-H-6083C	MIL-H-5606B	MIL-L-6081C	0.8567
MIL-H-6081C	MIL-H-6083C	MIL-L-17672B	0.8632
VV-I-530a*	VV-D-001078 (10 cs)	VV-I-530a	0.8700
MIL-L-17672B (high)	MIL-S-21568A (1 cs)	VV-D-001078 (10 cs)	0.9239

* Except less viscous than 6081C at 20,000 psig.

3.6.6 Viscosity Ranking

Viscosity Ranking	Kinematic Viscosity cs at 100° F 0 psig
MIL-S-21568A (1 cs) GE(TM) SF-96-1.0 (General Electric Co.) DC(TM) 200-1 (Dow Corning) GE(TM) SF-1143-1 (contains lubricity additive)	0.76 (1.0 at 77° F)
DC(TM) 200-1.5	
DC(TM) 200-2.0	
DC(TM) 200-3.0	
MIL-H-46004	2.88
Royal(TM) C-141	3.08
Brayco(TM) Micronic 713 (Bray Oil Co.)	3.42
Brayco(TM) Micronic 762	3.73
Hoover(TM) Submersible 2 (Hoover Electric Co.)	4.26
GE(TM) SF-96-5.0 DC(TM) 200-5.0	(5.0 at 77° F)
Shell(TM) Tellus 11	4.68
MIL-H-81019B	7.20
Marcol(TM) 52 (Humble Oil & Refining Co.)	7.68
VV-D-001078 (10 cs) DC(TM) 200-10 GE(TM) SF-96-10.0	8.66 (10.0 at 77° F)
VV-I-530a, Type 1 (Esso(TM) Univolt 33) Type 2 (Esso(TM) Univolt 34)	9.65
MIL-L-6081C, Grade 1010	10.08
MIL-L-7870A	10.30
Shell(TM) Tellus 15	10.80
MIL-H-6083C (high shear)	11.78
MIL-H-5606B (high shear)	12.26
MIL-L-6085A	12.70
MIL-H-5606B (low shear)	13.80
MIL-C-8188C	14.14
MIL-H-27601A	15.11
MIL-H-6083C (low shear)	15.80
Brayco(TM) 783	16.00
MIL-L-7808G	17.30
New(TM) Departure Hyatt NDH-TD4-1	19.7
MIL-L-23699A	25.67
MIL-F-17111	28.80
Shell(TM) Tellus 27	34.00
MIL-L-17672B, MS 2110TH	35.68
Houghton(TM) PR-1192	41.90
Primol(TM) 207 (Humble Oil & Refining Co.)	44.10
VV-D-001078 (50 cs)	(50 at 77° F)
Houghton(TM) PR-85-29-129	67.30

(TM) Trade names marked with this symbol are proprietary to the manufacturer.

Note that the above ranking of fluids was according to basic viscosity data only. When viscosity characteristics are considered for a given range of temperature and pressure, the comparative ranking of fluids will likely be altered. For example, at 35° F, MIL-H-6081C and VV-I-530a are considerably more viscous than either 5606B or 6083C; but at 100° to 150° F., the viscosity data for all four fluids are quite comparable from 0 to 10,000 psig (see section 3.6.2). For fluid-filled machinery applications, the fluid selected must provide satisfactory lubrication for surface or near-surface operation and yet must not be so viscous at depth that power losses are excessive.

3.6.7 Flash Point Ranking

Flash Point Ranking	Flash Point, ° F, per ASTM D-92 Method[23]
GE(TM) SF-96-1.0 (General Electric Co.)	110 (manufacturer's data)
DC(TM) 200-1 (Dow Corning)	110 (manufacturer's data)
GE(TM) SF-1143-1	110 (estimated)
MIL-S-21568A (1 cs)	115
MIL-J-5624F, JP-5	140 min.
DC(TM) 200-1.5	145 (manufacturer's data)
DC(TM) 200-2.0	175 (manufacturer's data)
Hoover(TM) Subm. No. 2	185
Brayco(TM) Micr. 762	200
Brayco(TM) Micr. 713	205
MIL-H-46004	210
MIL-H-81019B	212
DC(TM) 200-3.0	215 (manufacturer's data)
Royal(TM) C-141	215
MIL-H-5606B	215
MIL-F-17111	>220
MIL-H-6083C	230
Houghton(TM) PR-85-29-129	265
New(TM) Departure - Hyatt NDH-TD4-1	270
GE(TM) SF-96-5.0	275 (manufacturer's data)
DC(TM) 200-5.0	275 (manufacturer's data)
Shell(TM) Tellus 11	280
MIL-L-7870A	285
Shell(TM) Tellus 15	300
MIL-L-6081C, Grade 1010	305
DC(TM) 200-10	325 (manufacturer's data)
VV-I-530a	325
Marcol(TM) 52 (Humble Oil & Refining Co.)	330
VV-D-001078 (10 cs)	355
MIL-L-17672B, MS 2110[illegible]	360
Houghton(TM) PR-1192	375
MIL-L-6085A	385
MIL-H-27601A	390
Shell(TM) Tellus 27	395
Primol(TM) 207 (Humble Oil & Refining Co.)	400
GE(TM) SF-96-10	410 (manufacturer's data)
MIL-L-78086	415
MIL-C-8188C	455
MIL-L-23699A	490
VV-D-001078 (50 cs)	>535

3.6.8 Cost Ranking

Cost Ranking	$/Gallon (approx)
MIL-L-6081C, Grade 1010	0.72
VV-I-530a Type 1, Esso™ Univolt 33 Type 2, Esso™ Univolt 34	 0.73 0.77
MIL-L-7870A	0.76
Shell™ Tellus 11, 15	0.83
Marcol™ 52 (Humble Oil & Refining Co)	0.83
Shell™ Tellus 27	0.85
MIL-J-5624F, JP-5	1.00
MIL-H-46004	1.12
MIL-L-17672B, MS 2110TH,	
Houghton™ 681	1.16
MIL-H-81019B	1.27
Houghton™ PR-1192	1.34
Primol™ 207 (Humble Oil & Refining Co.)	1.36
MIL-H-5606B	1.42
MIL-H-6083C	1.62
MIL-L-6085A	2.00
MIL-F-17111	2.22
Brayco™ Micronic 713 (Bray Oil Co.)	3.56
Brayco™ Micronic 762	3.56
MIL-L-23699A	4.70
MIL-C-8188C	5.[illegible]0
MIL-L-7808G	5.60

Cost Ranking	$/Gallon (approx)
Hoover™ Submersible 2 (Hoover Electric Co.)	6.08
VV-D-001078 (10 cs)	
DC™-200-10 (Dow Corning)	12.60 or 1.60/lb
DC™-200-5.0	12.60 or 1.60/lb
GE™-SF-96-10.0 (General Electric Co.)	1.80/lb
GE™-SF-96-5.0	1.80/lb
VV-D-001078 (50 cs)	15.00
MIL-S-21568A (1 cs)	
DC™-200-1	4.85/lb
GE™-SF-96-1.0	5.10/lb
DC™-200-1.5	
DC™-200-2.0	4.85/lb
DC™-200-3.0	
MIL-H-27601A	65.00
GE™-SF-1143-1 (1 cs)	70.00

3.6.9 Fluid Volume Thermal Expansion Coefficients

The data presented on this sheet are believed to be accurate, but are not to be construed as official or reflecting research data of the U.S. Navy.

Fluid Type	Temperature Range, ° F	Volume Thermal Coefficient of Expansion, cc/cc/° F
MIL-H-5606B Hydraulic, aircraft, red	0 300 60-160	0.00043 } one manufacturer 0.00530 } 0.00041 another manufacturer
MIL-H-6083C Storage, emulsifying, red	← Approximately same as for MIL-H-5606B →	
MIL-L-6081C Grade 1010 Turbine Oil	60-200	0.00046
Brayco™ Micronic 762	0-200	0.00044
Petroleum, 5 cs	100	0.00045
Silicone Fluids:		
DC™-200-1.0 (Dow Corning)*	-	0.000744
DC™-200-1.5	-	0.000744
DC™-200-2.0	-	0.000650
DC™-200-3.0	-	0.000589
DC™-200-5.0	-	0.000583
DC™-200-10.0	-	0.000600
DC™-200-20.0	-	0.000594
DC™-200-50.0	-	0.000578
GE™-SF-96-5.0 (General Electric Co.)	77-302	0.000583
GE™-SF-96-10.0	"	0.000600
GE™-SF-96-20	"	0.000594
GE™-SF-96-50	"	0.000589
GE™-SF-85 (20)	39-189	0.000516
GE™-SF-97 (20)	36-185	0.000530
GE™-SF-97 (20)	77-302	0.000594
GE™-SF-97 (50)	"	0.000589
GE™-SF-97 (100)	"	0.000514

*The Dow Corning DC-200 silicone fluids were designed to meet MIL-S-21568A. While this specification still applies to 1-cs dimethyl polysiloxane fluids, it has been superseded by Federal specification VV-D-001078 for dimethyl polysiloxane fluids ranging in viscosity from 0.65 to 200,000 cs.[23]

Note: Even for specification fluids, these data may differ for each manufacturer's product.

CHAPTER IV

FLUID CONDITIONING, RECONDITIONING, AND FILL PROCEDURES

4.1 FLUID FILL PROCEDURES (As Applied To Individually Housed Submersible Electric-Drive Motors)

It is absolutely necessary to provide fluid ports at the lowest and highest points of each cavity in a drive assembly to facilitate fluid filling, venting, and draining. This includes not only the motor but also the speed reducer, interseal cavity, and any integral compensating devices.

The following approaches have been used for filling motors:

- Gravity.
- Forced or pressurized.
- Vacuum.

Although somewhat elaborate and time consuming, vacuum filling of electric drive motors is the more conservative filling method. Vacuum degassing and filtering of the compensating fluid are desirable for any approach.

Generally, fill procedures have several objectives. First and most important, is to remove all air and fill all voids in the unit. Most compensating systems are sized to make up the volume lost by compression of the fluid at high pressures, usually allowing for a small amount of trapped and dissolved air. Since the compressibility of air is many times that of fluid, any significant amount of air left in the unit could cause the compensator to deplete prematurely thus collapsing the case. The second important reason for removing all air is heat transfer. All fluid-filled motors are designed to operate completely flooded which allows them to operate at higher power levels than they could operate at in air. If an air pocket is left in the motor it could cause overheating in the windings with subsequent insulation failure and winding burnout. It is also for this reason that any gas generation in the motor can have catastrophic effects. Another reason for removing trapped or dissolved air from the motor is the possibility of foaming in some fluids under certain conditions. Foaming could cause displacement of fluid from the motor and also excessive drag losses.

The second objective of the fill procedure is to set the compensator at its correct starting position. This position is usually determined by the manufacturer after considering the following factors:

- Thermal expansion of the fluid.
- Compressibility of the fluid.
- Expected seal leakage.
- Gassing.
- Safety factor.

The compensator position is usually set about 2/3 to 3/4 full. This allows for thermal expansion at the surface (which can be considerable) at the start of operation. One design employs a relief valve between the motor cavity and compensator cavity. The unit is filled without allowing any volume for thermal expansion. As the fluid heats up, the pressure in the motor increases until the relief valve opens and dumps some of the oil into the secondary cavity. This appears to be a good approach as it is often a difficult procedure to determine when the compensator is in its correct starting position.

The third objective of the fill procedure is to remove all dissolved oxygen from the fluid. Tests conducted at this laboratory have shown that corrosion due to sea-water leakage into the unit will be drastically reduced if oxygen is not present.

The fourth objective is to filter the fluid just prior to putting it into the unit without exposing it to possible sources of contamination.

For gravity filling of a motor, fluid is slowly admitted through the upper, vent port until the housing is full. The motor may be tilted, shaken, and the shaft turned by hand to aid in air removal and the penetration of fluid into all voids. It is also beneficial to allow the motor to stand for periods of time and to repeat the above process. When the fluid level no longer needs to be "topped-off," the fill port may be sealed.

In forced or pressurized filling, fluid is admitted to the lower, fill/drain port of each cavity while the vent port is opened. The fluid flow rate should be very low. Normally a 5- to 10-micron filter is included in the pressurized fill line to ensure that no contaminants will be introduced from the filling apparatus. It is

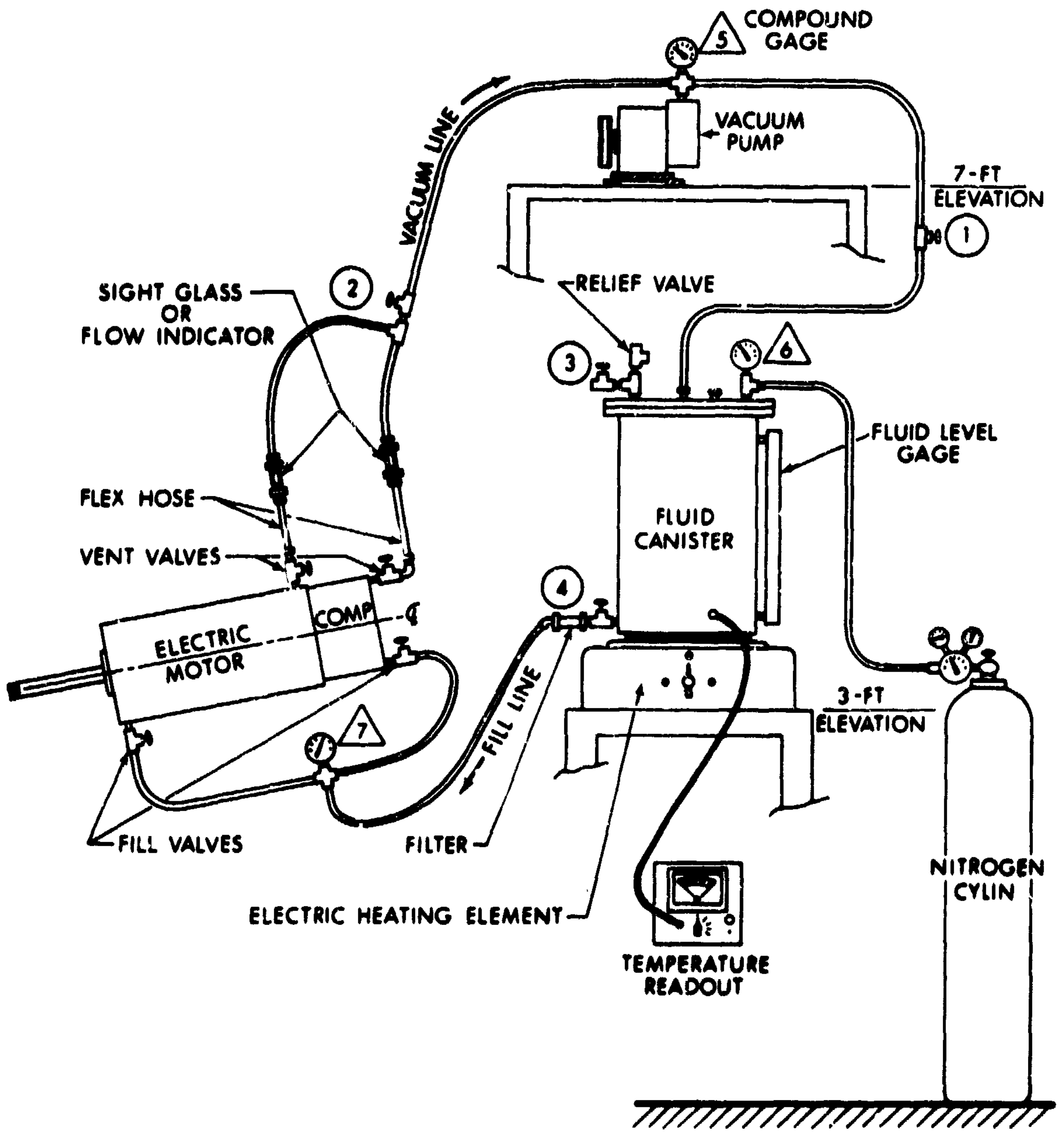
desirable to use degassed fluid and to supply a filling pressure head with an inert gas such as nitrogen. This will minimize entrained air in the incoming compensating fluid. Although the fluid may be recirculated several times, there is no assurance that entrapped air will not remain in the motor case.

Thus, an improvement to a pressurized fill process is to maintain a preselected vacuum on the vent port and internal motor case to remove all air from the unit. A precaution is that the amount of vacuum used should not be injurious to shaft seals or compensator elastomers (such as rolling diaphragms). This will enable the incoming fluid to thoroughly fill voids in the drive machinery.

It is advantageous to slightly pressurize a filled drive assembly (say, 3-5 psi), then vent and top-off each cavity. To force fluid into any remaining voids, high ambient pressure cycling may be used. A fluid-filled drive system should be subjected to at least one and preferably three pressure cycles to rated operational pressure. Then the fluid level should be checked and topped-off once again. The following vacuum fill procedure and explanation will further clarify this fluid-filling rationale.

4.2 ANNAPOLIS FILL PROCEDURE

Refer to following schematic and description of the components in the fill system.



Vacuum Fill System for Submersible Electric Drive Equipment

Description of Vacuum Fill Components

1) Valves 1 through 4. Whitey Type 1KF4, brass, 1/4 NPTF ends, KEL-F valve stem tip especially for vacuum applications. Do not over-torque valve when closing - will eventually damage KEL-F tip. Whitey Research Tool Company, Emeryville, California.

2) Compound Gages, items 5, 6, and 7. 30 in. Hg. vacuum to 30 psi. Marsh Instrument Company, Skokie, Illinois.

3) Flow Indicators. Model F1GG-3/8 D1. Republic Manufacturing Company, Cleveland, Ohio.

4) Liquid Level Gage (on canister). Oil-Rite Corporation, Manitowoc, Wisconsin.

5) Filter, brass body, type 316 sintered stainless steel filter element, nominal 7-micron filtration, Nupro Series 6F-7, Nupro Company, Cleveland, Ohio.

6) Vacuum Pump, Duo-Seal, W. M. Welch Manufacturing Company, Chicago, Illinois.

7) Flexible hose, medium pressure, Aeroquip Corporation, Jackson, Michigan.

8) Fittings: Both 1/4 inch flared tubing connectors and 1/4-inch "Swagelok"(TM) fittings have been used to connect flexible hose or tubing to motors. "Swagelok" - Crawford Fitting Company, Solon, Ohio.

Note: A primary consideration in the choice of the above components was for vacuum service to 28 in. Hg.

a. For a fluid such as MIL-H-5606B, vacuum degas fluid under the following conditions:

(1) Pull vacuum from 24-28 in. Hg, if possible, on volume of fluid within sealed canister.

(2) Heat fluid to 150° F (175° F maximum) to aid in the removal of entrained air in the fluid. It is felt that this procedure will not damage the fluid, since a sample of MIL-H-5606B was heated in excess of 200° F and yet the fluid electrical properties were essentially unchanged. At atmospheric conditions, fluids have a natural tendency to absorb ambient air and to hold it in suspension (see section 4.6). Vacuum degassing will put the fluid in an unsaturated, unstable condition.

(3) As the fluid is being heated, it will be necessary to run the vacuum pump intermittently to maintain 24-28 in. Hg. Continue this process for at least several hours.

(4) For very thorough degassing, it is advisable to stir the fluid by some means. Ordinarily this was found to be impractical. A strong magnetic stirrer may be feasible.

(5) Bubbles will appear to rise in the canister fluid level gage as the degassing process continues.

b. Set up vacuum fill apparatus for the subject motor, with or without compensator as shown in the schematic.

c. From vacuum degassing, valves 2, 3, and 4 and the nitrogen cylinder valve should be closed. Valve 1 was left open in order to pull a vacuum on the fluid in the canister.

d. Position motor so that the vent or vacuum ports of each cavity are at the highest point. Fluid should enter the lowest point in each cavity, if possible. When necessary, motor may be tilted to achieve this. Attach fill and vent hoses to motor as shown. Note valves, sight glasses in vent lines, and compound gage in fill line.

e. Proceed to close valve 1 and open valve 2. Open the fill and vent valves to the motor. Start vacuum pump and run for several minutes to evacuate air from motor and compensator cavities.

f. Pull 26-28 in. Hg. vacuum on the motor or whatever is feasible, considering shaft seal leakage or manufacturer's specifications for maximum vacuum on compensator, etc. When the pump is stopped, note whether the vacuum holds or tends to "fall off." For the vacuum fill process, try to maintain what vacuum is feasible for the particular motor/compensator unit.

g. Open the nitrogen cylinder valve to pressurize the canister to 15 psi. Other inert gases may be used to supply a pressure head to the fluid. <u>The main consideration is to avoid reintroducing oxygen into the degassed fluid.</u> For a 15 psi head, 5-8 psi may be required to overcome the pressure drop of the nominal 5- to 10-micron filter. A fluid temperature of 130°-150° F will decrease the viscosity and thus aid in the flow rate

through the filter and then into the smaller voids of the motor.

h. To maintain a vacuum, run the pump as necessary during the fill process. When fluid appears in the sight glasses, stop the fill process (close valve 4). Keep pulling a vacuum until no bubbles are seen rising through the flow indicators (visual sight oil traps may also be used). The motor may be shaken to aid in the removal of bubbles in the fluid. It may also be necessary to close valve 2 to prevent oil from entering the vacuum pump. The pump oil case may be drained if contamination occurs and replenished with vacuum pump oil.

i. Stop vacuum pump. First close valve 2 and then close vent valves on motor. Open valve 4 and overpressure motor 3-5 psi (read gage 7) to force oil into any remaining voids. Close valve 4.

j. After at least one hour at 3-5 psi overpressure, close fill valves to motor and disconnect fill and vent hoses. Relieve motor vent valves to ambient pressure. Allow motor to stand for several hours, then shake motor and check fluid level. Top-off if necessary. Repeat, if necessary, so that fluid level in motor does not significantly decrease.

k. Adjust compensator to recommended position. Top-off motor and compensator cavities before plugging ports.

l. Compensator and/or motor fluid level should be checked after first pressure cycle. Some decrease in fluid volume, even with no apparent leakage, seems to be a typical phenomenon.

4.3 FILL CONSIDERATIONS

It is important to keep in mind that the filling procedure must be carefully planned for each individual unit, considering the following.

The number of fill and vent ports for each cavity and their respective locations. We prefer that the vent port be located at the highest point of the cavity and fill port at the lowest. This is usually not the case and the motor must be oriented with the vacuum line at the highest point and the fill line at some point in between. Often times the cavity has only one port. This requires that the vacuum and fill be through that port with a different valving arrangement. First the vacuum is pulled on the cavity. Then a valve in

the vacuum line is closed and a valve in the fill line opened allowing fluid to fill the evacuated cavity. This is not considered a good procedure; it is highly recommended, therefore, that all cavities have two holes so that simultaneous evacuation and filling can be used.

A major problem is encountered when two cavities must be filled simultaneously and the highest point on one cavity does not coincide with the highest point on the other. In this case vacuum filling may not be very effective.

Several instances have been encountered where it was necessary to pull a vacuum on two or more cavities simultaneously. In one case it was because of a carbon face seal between the two cavities. The vacuum would have been working against the spring and could have dislodged the seal. In another case there was a rolling rubber diaphragm between the compensator and motor cavities. The compensator was designed so that the pressure inside the motor is always higher than the pressure in the compensator cavity. The diaphragm may be subjected to a pressure differential only in that direction. If a vacuum were pulled on the motor cavity and the compensator left at ambient pressure, the diaphragm would be pulled out of shape and not operate properly.

When two cavities are filled simultaneously the routing of the vacuum lines becomes important, since one cavity usually fills before the other and oil can flow into the sight glass giving a false fill indication.

The operation of O-ring seals is somewhat questionable during this process. First the O-rings are seated on one side of the groove by a vacuum in the motor, and then they are forced over to the other side of the groove by the slight overpressure in the motor cavity during actual operation. In fact, the entire technology of low pressure differential seals at high ambient pressures needs to be investigated.

One final word of caution. Although every precaution is taken, the unit may still have some trapped air. As a final step before energizing the system under pressure, the unit may be subjected to three pressure cycles to rated pressure while monitoring differential pressure between the motor cavity and ambient pressure. If the pressure inside the motor becomes less than the external ambient pressure, stop the test. This usually means one of four things:

- There is a leak in the system which has depleted the compensator.

- Because of the extremely high pressure, the oil is forced into voids which could not be filled at low pressure.

- Air is present in the motor.

- The compensator is undersized.

Upon removal of the unit from the pressure tank, check the condition of the compensator. If the compensator has returned to its initial position, air is probably trapped in the system. If the compensator has become depleted the unit should be carefully checked for signs of oil leaks. If none can be found, then replenish the depleted oil and repeat the pressure test. This may be repeated several times before an equilibrium condition is reached. When the internal fluid pressure never falls below ambient, and when subsequent inspection reveals that the compensator returns to its original position, the unit is considered ready to be energized and tested.

It should be pointed out that in applications where several hundred hours of unattended operation are required, even a minimal leakage rate cannot be tolerated.

4.4 COMPENSATOR FILL PROCEDURES

If the compensator is integral with a motor/speed reducer housing, vacuum filling is desirable to more thoroughly fill voids in the assembly with degassed compensating fluid. However, when a vacuum-fill procedure is devised, careful consideration must be given for various physical aspects of the drive system:

- Housing geometry-cylindrical and spherical housings can withstand up to 28 in. Hg vacuum, whereas "box" enclosures can only withstand 1 to 2 in. Hg vacuum without excessive structural stiffening.

- Static and dynamic seals: Will any seals be unseated, "sucked-in," or deformed under vacuum?

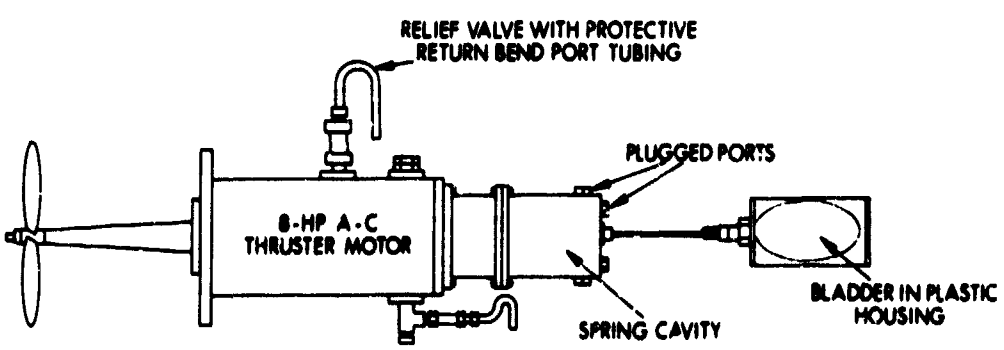
- Compensator elastomer: What amount of vacuum is reasonable so that a rolling diaphragm or other elastomeric membrane will not extrude, rupture or excessively deform?

In reference to vacuum filling the rolling diaphragm motor compensator in section 3.5.2, it will be necessary to maintain a vacuum on both the inboard fluid cavity and the outboard, normally sea-water-flooded,

spring cavity to protect the elastomeric diaphragm from inverting and rupturing (refer to the section drawing of the compensator). The indicator-guide rod penetration and the sea-water filtering ports must both be sealed and a vacuum line attached to the outboard spring cavity.

The spring bias will "bottom-out" the diaphragm so that the compensator is initially at minimum stroke for the vacuum fill process. After the motor cavity is full of fluid, remove the vacuum lines from the motor cavity and outboard spring cavity and insert the indicator-guide rod. Install check valves 1 and 2 as shown in drawing. Pressurize the fill fluid to about 1.5 psig at room temperature. This will force in additional fluid to displace the rolling diaphragm from minimum to a mid-travel position. The indicator-guide rod provides a means to visually monitor the compensator position. Install the sea-water filters in the ports of the spring cavity and the motor will then be ready for service.

For added protection, the compensation can be cascaded. Rather than to expose the spring cavity and rolling diaphragm directly to sea water, the spring cavity may be vacuum or gravity filled with degassed fluid. Then attach a fluid-filled bladder or other compensator as shown below.



Cascaded Compensation

The feasibility of attaching an additional compensator is a limitation of this cascaded approach.

Compensators may be vacuum filled and pressurized, but often the most feasible approach is to simply gravity fill and thoroughly bleed the device. For filling

elastomeric bladders and tubes, one technique is to pull a slight vacuum and then admit degassed fluid at a low flow rate. Otherwise gravity filling in air must be done quite slowly to minimize air entrainment. When only one port is available, as for in-air gravity filling of a bladder, locate the port vertically upright and employ a slow "trickle-fill" procedure.

Any compensating device should be tilted, inverted, or otherwise manipulated or positioned to best facilitate fluid filling and bleeding. The desired situation is to connect a fluid fill line to the lowest port while the vent or bleed port should be at the highest point. Agitating, shaking, massaging compensator elastomeric materials and pressurizing may be employed to aid in thorough filling. The compensator assembly may also be allowed to rest for a period of time and then "topped-off" with fluid.

Dissolved air and gases in compensating fluids are noncompressible as long as they remain in solution. Air trapped by fluid during the fill process is referred to as free air. Air which comes out of solution may coalesce and become entrained in the system. Entrained and free air in a fluid system are compressible. To ascertain excessive air inclusion, inadequate compensator volume allowance, or fluid leakage problems, the compensated drive system may be subjected to one or more pressure cycles in a hydrostatic pressure tank. Pressure cycling may force air or gas into higher portions of the compensator and drive machinery. Bleed and top-off the system as necessary (see following test methods).

4.5 COMPENSATOR TEST METHODS

- A filled compensating system should be visually inspected to determine any leaks. For locating even negligible fluid seepage, a black light has been found to be quite effective.

- Provided the structural design and elastomeric materials can withstand a certain proof pressure, fluid overpressurizing for a short period of time is also a good technique for locating leaks.

- A rigorous test method is to subject the compensated system to operational (rated) ambient pressure or 1.5 x operational pressure (test depth) for at least one and preferably three pressure cycles. To simulate severe environmental conditions, cool the water in the hydrostatic test tank to the lowest forseeable

temperature and then subject the cold drive system to maximum ambient pressure. It is advisable to monitor compensator volume travel or differential pressure during pressure cycling. If for any reason the compensator volume is depleted, the equipment housing will implode or otherwise fail.

4.6 FLUID CONDITIONING

The importance of vacuum degassing a compensating fluid cannot be too strongly emphasized. Since fluids have an attraction for air, at standard room conditions it will be absorbed and dissolved to a percentage of volume which is referred to as "saturation level." The natural dissolved air content in fluids ranges up to 14% by volume. Air which is in solution with a fluid (dissolved air) has no effect on bulk modulus, whereas <u>only 1% by volume free air or entrained air decreases bulk modulus more than 50%</u>.

- Free air is that which is entrapped in the compensating system, as in voids or air pockets. Although trapped, it is not totally in contact with the compensating fluid. Normally free air is removed by vacuum or by thorough <u>bleeding</u>.

- Entrained air in a fluid usually exists in the form of small bubbles. It is in suspension and can be removed by filters which have a low bubble point.

- Dissolved air or gas enters into solution with the fluid and can come out of solution under a vacuum situation such as occurs in throttling across an orifice.

When dissolved air comes out of solution, it can be partially readsorbed by the fluid as long as it is in bubble form (entrained). However, as bubbles collect and coalesce, free air pockets are formed and fluid compressibility increases. At an ambient pressure of 500 psig, significant changes in compensating fluid volume will occur when only small amounts of free or entrained air are present. Neither the absence nor the presence of dissolved air affects the volume of the fluid. Only the weight of the fluid will change, but not the volume.

Thus, air or gas should be removed by subjecting a compensating fluid to a vacuum and/or heating. Henry's law states that the weight of gas dissolved in a liquid is proportional to the pressure of the gas. Not only does the solubility of a gas in fluid decrease as

pressure is reduced, but an increase in fluid temperature also tends to liberate more dissolved gas. Vacuum degassing minimizes fluid compressibility and places the fluid in an unsaturated, unstable, yet desirable condition. After a drive system is vacuum filled, degassed fluid acts as a sponge to absorb any trace amounts of air into solution.

For vacuum degassing a fluid such as MIL-H-5606B, refer to the previous section 4.2 and review the fill procedure. Although this approach is not elaborate and has some limitations, it provides basic guidelines for conditioning fluids. In the vacuum fill procedure note that an inert gas is used to supply the pressure head for final-filtering the degassed fluid.

In conditioning a compensating fluid, there is a need to determine certain "levels of quality" which can be tolerated in an operational system. Periodic fluid samples should be taken and evaluated to measure the following:

- Air inclusion: Is this level acceptable for system requirements and the volume capacity of the compensator?

  Considerations: Maintain dissolved air content to a point below the natural saturation level of the fluid. A limiting factor in degassing is that the desired vacuum level at a particular temperature must be higher than the vapor pressure of the fluid. If the absolute pressure over the fluid is less than the vapor pressure of the fluid, then the fluid will begin to boil.

- Cleanliness level: Determine the presence of entrained or dissolved water, acids, and solid contaminants.

  Considerations: The filtering medium and degree of filtration must remove various contaminants and yet preserve the critical properties of the fluid. For adequate solids removal and purification, several cycles of filtering and heating under vacuum may be required (reconditioning).

- Dielectric quality: Possible degradation in values of insulation resistance, dielectric breakdown voltage, dissipation factor, and resistivity.

  Considerations: Minimum acceptable dielectric values must be established depending upon the specific electrical requirements of the motor

or controller. Measurements should be made when no water or solids are present in the fluid or after a satisfactory cleanliness level has been obtained by reconditioning a used fluid. If measured electrical properties are marginal or below minimum acceptable values, a decision must be made either to discard the fluid or to submit it to a commercial reclaiming process. Unless the fluid is quite expensive, the cost involved in commercial reclaiming is usually prohibitive for the degree of rejuvenation that would be obtainable.

4.7 MECHANICS OF FLUID RECONDITIONING

4.7.1 Sampling

A minimum 100-ml fluid sample should be taken from the collection sump or drain of the motor, pressure-compensated enclosure, etc. If the initial fluid appears emulsified or cloudy, this indicates sea water and/or solids contamination. Samples which have a poor visual appearance, due to the presence of decomposition products, may have relatively good dielectric properties. Hence, a periodic measurement of dielectric properties is the more valid approach.

4.7.2 Fluid Maintenance

Normally, fluid systems on a deep submersible are checked out "predive" and "postdive." Significant sea-water intrusion will be indicated by sea-water leakage detectors in the system. An optimum frequency of fluid maintenance will be determined by general operating experience. For instance, if operating temperatures are low, a regular change interval may not be necessary from a fluid life standpoint. However, the drive machinery fluid may need to be drained and reconditioned after every three 8-hour missions to maintain the desired "cleanliness level."

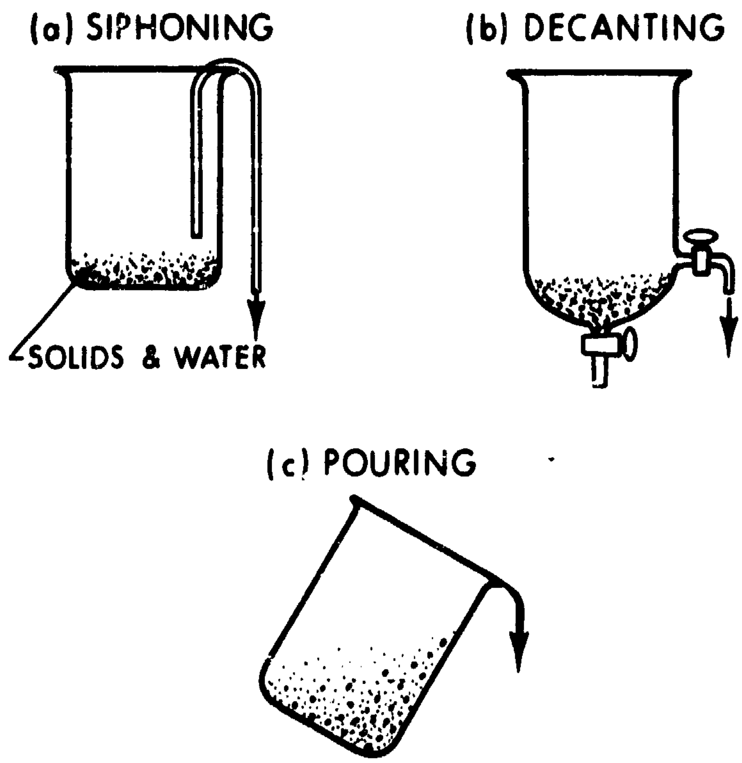
When it is necessary to drain the fluid, discard initial concentrations of sea water and solid matter and then drain the remaining fluid. Thoroughly flush the particular housing or compartment, with a recommended flushing procedure. It is imperative to remove all remaining gum, sludge, solid products, and sea water.

A cleaning or flushing fluid must be compatible with the compensating fluid so that no insoluble precipitates or residue will be formed. One flushing procedure employs MIL-H-6083C as a flushing oil for MIL-H-5606B (see section 3.3.2).

Extreme care must be exercised in using solvents for cleaning and flushing drive systems and compensators. From a materials compatibility standpoint, solvents will soften and deteriorate insulation systems and elastomers. Moreover, even trace amounts of various alcohols, methyl ethyl ketone, acetone, etc in a compensating fluid will reduce dielectric properties. Solvent-cleaned parts must be thoroughly disassembled and dried to preclude these problems.

4.7.3 Gravity Settling

If time permits, gravity settling is effective for the removal of gross contamination. It is best for the fluid to remain quiescent. (In point of fact, this is not practical for fluid reconditioning on board a support ship.) As a general minimum time period, allow the fluid to stand overnight. Finer dispersions of contaminants require longer settling time. Fluids having both a higher viscosity and a higher specific gravity will have slower settling rates. Three means of recovering settled fluids are shown below. Of these, pouring off the clean fluid is the least desirable approach, since agitation of the contaminants is more likely.



Settled Fluid Recovery Techniques

Also, the higher the interfacial tension between the fluid and entrained particulates and water, the more the settling rate is retarded. Fluids which contain emulsifier and/or corrosion-inhibitor additives can form stable emulsions with dissolved water. Emulsification characteristically produces a cloudy or "milky" appearance in a fluid. MIL-H-6083C hydraulic fluid contains a sulfonate additive which both increases its ability to emulsify sea water and acts as a corrosion inhibitor. This fluid, under strong agitation, will emulsify up to 10% sea water by volume. Even after long-term settling, an emulsion of some extent will remain. A solvent or other liquid which could effectively lower viscosity and break the emulsion would likely affect lubricity and chemical properties of the fluid and reduce dielectric properties. To date, no simple procedure has been devised to thoroughly recondition an emulsified fluid and such a process may not be economically worthwhile.

4.7.4 Filtering and Water Removal

Fluids have a natural tendency to become saturated with air and water. This tendency increases with pressure. At the ocean surface, after a submerged mission, sea-water intrusion in nonemulsifying compensating fluids will tend to come out of solution and coalesce at the reduced ambient pressure. If water or solid matter are present in the initially drained fluid or if a fluid sample appears cloudy, indicating suspended contaminants, throughly drain all fluid. The consideration must be weighed whether to allow fluid contamination to settle out at least overnight or to proceed directly with filtering out contaminants. Many filtering schemes may be feasible, but usually the range of filtration is from 5 to 10 microns.

Filter/separators have been effective for removing particulates and free water from various fuels and fluids. These are essentially two-stage devices, usually having two different types of filter cartridges. Whereas a single low micron filter may readily become clogged, the first-stage cartridge of a filter/separator has the ability to remove larger particulates and coalesce entrained water. Water droplets settle to the bottom of the filter housing where they may be drained. Second-stage elements are composed of hydrophobic materials which separate any remaining water and filter out smaller particulates. This method of water separation is dependent upon a high interfacial tension between the fluid and the water. Optimum conditions must be maintained to ensure contaminant removal. Thus, type and concentration of contamination and the susceptibility

of the filtering element becoming saturated directly influence the effectiveness of filter/separator fluid reconditioning. Filter/separators will also remove certain additives from petroleum-base fluids.

Fuller's earth and other clay type filters accomplish contaminant removal through an adsorption process over the large surface area of a cartridge. A means must be devised to check the performance of a this type filter, since a portion of the unused adsorptive area must remain available to filter the fluid. There is evidence that Fuller's earth filtration may also affect or remove additives in compensating fluids. Some salt content from sea-water intrusion may also be removed. To reduce the sea-water emulsibility of a fluid, as an aid in water removal, this type of filtering medium may be desirable. However, when filtering a heavily contaminated fluid, it is possible to saturate and "disarm" the clay cartridge so that there is a continuous penetration of finely dispersed water. There is no simple way to determine when the element has become saturated. Fuller's earth filters have been rejuvenated by heating under vacuum or by simply "baking out" the water. But additives and salt will remain in the filter cartridge.

Sintered stainless steel filter elements are effective for removing solid contaminants, but a large effective surface area must be available to achieve reasonable flow rates. High porosity membrane filters provide good filtering flow rates and good retention of suspended matter. Several polymer filter materials are available to provide optimum compatibility with the compensating fluid and contaminants. Membrane filters are available which have a hydrophobic property, that is, the filter surface resists wetting by water. This characteristic will enable a membrane to separate and retain suspended water in the fluid. Dissolved water, however, will not be filtered out of solution.

> Note: Water which is in solution with a compensating fluid (dissolved) will not be removed by a filtering process alone. At standard conditions, a fluid will likely have much less than 1% water by volume actually in solution. Greater amounts of sea-water intrusion in a fluid will exist in other forms, such as entrained or suspended, emulsified, or free water.

A recommended procedure is to use a depth-type prefilter so that a larger percentage of suspended matter

will be removed and not rapidly clog the downstream low-micron filter. For filtering very low viscosity fluids, it is feasible to use 3- or even 1-micron final filtration. High levels of fluid cleanliness are necessary for devices such as electrical contactors. ANNADIV NAVSHIPRANDCEN letter report ELECLAB 238/68[31] has been included as an example procedure in reconditioning low viscosity silicone fluids. The test method for measuring fluid electrical properties was per ASTM standards D1169-64 and D877-67.

4.7.5 Silicone Fluid Reconditioning Procedure

DEPARTMENT OF THE NAVY
NAVAL SHIP RESEARCH AND DEVELOPMENT CENTER
BETHESDA, MARYLAND 20034

IN REPLY REFER TO

C O P Y

NP/3960(A612 WEP)
Assigt 611 118
ELECLAB 238/68

14 NOV 1968

FOR OFFICIAL USE ONLY

From: Officer in Charge
To: Commander, Naval Ship Engineering Center (SEC 6156C)

Subj: ELECLAB 238/68, Quality Control Procedures for General Electric Co. SF 96-1 Silicone Fluid Used as a Compensating Fluid on Navy Submersibles

ABSTRACT

Several Navy Deep Diving Submersibles, including the Trieste II, utilize silicone fluids for pressure compensation, and lubrication in submersible motors, and as dielectric fluid surrounding electrical contactors. A procedure was needed to determine and maintain the quality of these fluids.

Samples of the General Electric Company's SF 96-1 silicone fluid, which is used on Trieste II, were evaluated for their dielectric quality. The samples included new, used, and reclaimed fluids. Recommendations made for quality control of SF 96-1 fluids follow:

1. Periodic sampling of the fluids to determine the presence of solids and/or water, and to measure electrical resistance and dielectric breakdown voltage.

2. Removal of solids and water, as necessary, by periodic filtration and settling.

3. Discarding or reclaiming of unsatisfactory material, as indicated by recommended procedures.

1. Introduction

General Electric Company's SF 96-1, a 1-centistoke dimethyl polysiloxane (silicone) fluid, is used on Trieste II as a pressure-compensating fluid for electrical components such as contactors. Where lubricity is also required, i.e., for use with electric motors, the SF 96-1 is modified by the addition of 5% (by volume) of dioctyl tetrachlorophthalate.

Because the SF 96-1 fluid, both straight and as modified for lubricity, is subject to degradation in use, and because of its high price (approximately $40 per gallon), either replacement or maintenance is an expensive procedure. Current practice is to have the used material periodically reclaimed by a commercial firm, at about half the cost of new fluid.

This Laboratory examined various samples of SF 96-1 provided by Lt. C. M. Staehle, of Trieste II, to determine if there were any appropriate procedures that could be recommended to help to improve maintenance and reduce costs. A specific point raised was whether periodic filtration of the fluid, using, for example, a 10-micron filter, followed by dehydration, might not be as effective as the commercial reclaiming process. The nature of the commercial reclaiming process is not known to this Laboratory, but is presumed to be distillation and/or chemical treatment.

Other questions to be answered appeared to be whether (1) an improved approach to the maintenance of the SF 96-1 fluids could be immediately suggested, and (2) the commercial reclaiming process is satisfactory in relation to its cost.

2. Approach

Various samples of SF 96-1 were tested as described below; the test results appear in Table I.

The procedures applied in this investigation included electrical measurements and special handling or processes.

a. Measurement of electrical properties

- The insulation resistance of all samples was determined with a General Radio type 1644A Megohm Bridge, using 1" diameter disc-type brass electrodes, spaced 0.100" apart.
- The specific resistance (resistivity) is calculated approximately by multiplying insulation resistance, in megohms, by 1.99×10^7, a conversion factor based on electrode area and spacing.
- The capacitance and dissipation factor were determined with a General Radio type 1611A Capacitance Test Bridge, with the same electrodes as used for insulation resistance.
- The dielectric breakdown voltage was measured with an Associated Research Inc. Model 4507 Oil Testing Hypot. ASTM procedure D877-67 was followed, except that the electrode spacing was 0.051"; the voltage rise rate was 600 volts per second; and five consecutive readings were taken on the same sample, with a 3 minute wait between readings. These deviations from the standard ASTM procedure have been found to yield more meaningful data for the present application than the standard procedure as written.

b. Handling of submitted samples (The designations of the various samples, written below in quotations, are the designations that were on the labels of the sample containers as received from Lt. Staehle.)

Sample No. 1 "New silicone oil," from Trieste II.

Since there was insufficient sample for running the tests, 50 ml. of this material was mixed with 50 ml. of fresh SF 96-1 from Annapolis stock, and tests were run on the mixture. In Table I, test results on straight SF 96-1 from Annapolis stock (sample 1a) are shown for comparison.

Sample No. 2 "Reclaimed silicone oil," from Trieste II

Tests were run on this sample as received.

Sample No. 3 "Contactor box silicone oil", from Trieste II

The sample was filtered through an 8-micron "Millipore" filter membrane prior to testing.

Sample No. 4 "Motor oil" (SF 96-1 + 5% by volume dioctyl tetrachlorophthalate), from Trieste II

The sample was filtered through an 8-micron "Millipore" filter membrane prior to testing.

TABLE I. TEST RESULTS ON SF 96-1 SILICONE SAMPLES

Sample	Appearance of sample as received	Temperature of sample °F	Insulation resistance, megohms	Approximate specific resistance (resistivity), ohm-cm	Dissipation factor, %	Dielectric breakdown voltage, kilovolts
1 "New ·ilicone oil" 50-50 mix with fresh SF 96-1 from Annapolis stock	Colorless, clear	76	4.1×10^4	8.2×10^{11}	0.9%	14.0 16.0 21.0 23.5 16.5 18.2 Average
1a Fresh SF 96-1 from Annapolis stock	Colorless, clear	75	2.5×10^4	5.0×10^{11}	1.0%	19.0 24.0 27.5 25.5 27.5 24.7 Average
2 "Reclaimed silicone oil"	Colorless, clear	77	3.5×10^5	7.0×10^{12}	1.2%	5.5 7.0 10.5 11.5 10.0 8.9 Average
3 "Contactor box silicone oil" (used)	Cloudy, discolored	78	2.6×10^5	5.2×10^{12}	1.1%	17.5 18.5 19.5 18.5 21.5 19.1 Average
4 "Motor oil" (used)	Cloudy, discolored	78	2.9×10^4	5.8×10^{11}	1.2%	19.0 21.0 21.5 22.0 19.5 20.6 Average

3. Comments on test results

a. Both of the samples of used material (samples 3 and 4) compare favorably with new SF 96-1. This is in agreement with earlier findings at this Laboratory that 1-centistoke silicone fluid does not undergo a rapid degradation in dielectric properties when subjected to prolonged repeated arcing.

The fact that these two samples were found to have relatively good dielectric properties, in spite of their poor visual appearance (due primarily to the presence of solid decomposition products), emphasizes the importance and value of periodic measurement of dielectric properties of the fluids, and the limitations of physical appearance as an indication of dielectric quality.

b. The reclaimed material is questionable because of its low dielectric breakdown voltage. The reason for the low values is not readily explainable. One possibility is that this particular lot had undergone degradation in dielectric properties as a result of extreme and prolonged usage prior to being reclaimed and was not improved by the reclaiming process.

4. General considerations relating to maintenance and possible rejuvenation of SF 96-1 fluid

Work previously undertaken, and continuing, at this Laboratory, indicates three possible causes of degradation of dielectric properties of SF 96-1 (and other fluids). The following paragraphs identify these three factors and explain the importance of each in the maintenance of SF 96-1 fluid. (The term SF 96-1 includes the "modified" fluid containing 5% dioctyl tetrachlorophthalate.)

a. The presence of suspended solids, particularly solid decomposition products, resulting from electric arcing in the fluid. The effect of this type of material on dielectric properties of the fluid has been found to be minor. Removal of solid products by filtration can by no means be considered an adequate method for rejuvenating the fluid. Filtration is, however, considered advisable to avoid build-ups that might contribute to bridging of gaps between conducting surfaces, or interfere with mechanical functioning of components.

b. The presence of suspended and dissolved water. Both suspended and dissolved water tend to lower the dielectric quality of SF 96-1. The effect of suspended water appears to be the more drastic of the two, and if the amount is excessive, short circuiting between conductive surfaces of components immersed in the fluid can occur. This effect can be readily reversed by periodic removal, by settling, of any suspended water that has accumulated. The removal of the additional dissolved moisture, such as by treatment with silica gel, is felt to be inadvisable, since the cost would be relatively high in relation to the small improvement obtained. Further, the fluid would probably become quickly re-saturated with moisture when put back into service, so that any improvement would be very temporary.

c. Chemical or electrochemical breakdown of the fluid. This is caused primarily by electric arcing in the fluid. It is not certain at this point whether "reclaiming", i.e., purification by distillation or chemical treatment, is feasible as a means of restoring dielectric properties to the original values of new fluid. However, as previously mentioned, the rate of degradation in dielectric properties of SF 96-1 due to chemical or electrochemical breakdown appears to be slow. In

the absence of reliable information from other sources, it will be necessary for Trieste II personnel to determine whether the present reclaiming practice is capable of restoring the dielectric properties of the fluid to their desired values, once these have dropped below the minimum limits of acceptability. Paragraph 5c identifies the necessary criteria.

5. Recommendations

The following steps are recommended for maintenance of SF 96-1 fluids on Trieste II. (The term "SF 96-1" includes both straight SF 96-1 and SF 96-1 containing 5% dioctyl tetrachlorophthalate.)

a. A minimum sample of 100 cc should be taken periodically from each separate pressure-compensating box, chamber, etc. where SF 96-1 is used. The sample should be taken in such a manner as to indicate whether there is any water or solid matter present on the bottom. The required frequency of sampling is to be determined by Trieste II personnel to insure that resistivity and dielectric breakdown voltage do not drop below the minimum acceptable values specified in 5e.

- If no water or solid matter is found on the bottom of the pressure compensating chamber, and if the SF 96-1 sample is clear, indicating no water or solid matter present in suspension, then resistivity and dielectric breakdown voltage should be measured, as outlined in 5d.

- If both values are above the minimum specified in 5e, the material can be considered satisfactory for continued service.

- If either value has reached the specified minimum, the material should be handled as outlined in 5c.

b. If the sample (paragraph 5a) is cloudy, indicating the presence of water and/or solids, or if water and/or solids are found on the bottom of the pressure compensating chamber, then all of the fluid should be drained out, including any water that may be present. The SF 96-1 should be filtered through a filter medium having a nominal pore size of 10 microns.

- If the filtered fluid is completely clear, its resistivity and dielectric breakdown strength can be measured immediately.

- If the fluid is cloudy, indicating suspended water, it should be allowed to settle overnight and the water removed by draining off the bottom, or by decanting the clear SF 96-1 fluid. Resistivity and dielectric breakdown strength are then to be measured.

- If the two dielectric properties are above the minimum values specified in 5e, the material can be considered satisfactory for further use. If either property has reached the specified minimum, the material is to be handled as outlined in 5c.

c. When either of the two dielectric properties of the fluids has reached the minimum values of 5e, a decision must be made either to discard the fluid or to submit it to the commercial reclaiming process. As stated in 3b, the value of the present reclaiming procedure is questioned, on the basis of test results on sample 2, Table I. The decision of whether to reclaim or discard from now on, should be determined by Trieste II personnel in the following manner: The next portion (or portions) of SF 96-1 that reach the minimum dielectric value(s) should be submitted

to the reclaiming process. If, after reclaiming, both properties have increased above the minimum values to the new levels shown below (when tested as described in 5d), the material is satisfactory for further use and the reclaiming process can be considered satisfactory and worth the cost involved.

MINIMUM VALUES
Resistivity at 65 - 85° F: 3.0×10^{11} ohm-cm
Dielectric breakdown voltage (average of 5 readings): 15.0 kilovolts

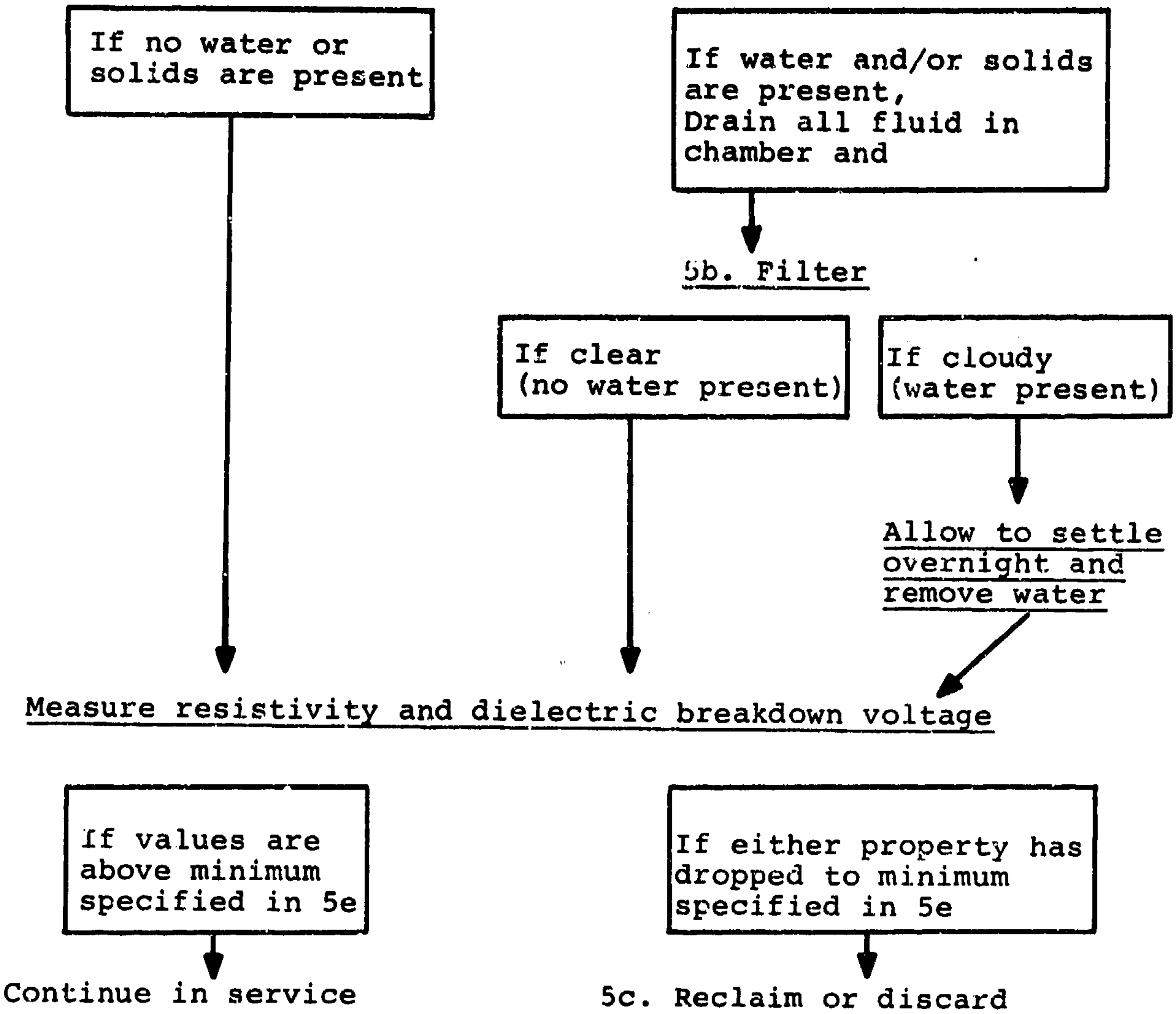
If either property is equal to, or less than, these values, the reclaiming process should be considered unsatisfactory, because the cost of reclaiming (approximately $20 per gallon) is felt to be excessive in relation to the degree of rejuvenation that would be obtained.

It should be noted that the portions of reclaimed material just referred to are in themselves satisfactory for some continued use (possible for only a short time) as long as their resistivity and dielectric breakdown voltage measurements are above the minimum values set forth in paragraph 5e.

For convenience, the recommended steps 5a, b, and c are given in outline form on the following page.

Summary of recommended maintenance procedure for SF 96-1 fluids

5a. Take minimum 100 cc sample. Look for water and/or solids in sample, or settled on bottom of chamber.



d. Recommended test procedures.

The two properties to be tested for maintenance of SF 96-1 fluids on the Trieste are resistivity and dielectric breakdown voltage. Dissipation factor, shown in Table I, is not considered a necessary measurement for maintenance purposes.

The recommended procedures are as follows:

(1) Resistivity. The method to be used is described in enclosure (1). The following suggestions are made for instrumentation:

Type 1644A Megohm Bridge, manufactured by
General Radio Co.
Concord, Mass.

This is a one-package instrument containing all of the required circuitry. The test cell must be purchased separately.

(2) Dielectric breakdown voltage. The method to be used is described in enclosure (2), with the following exceptions:

(a) Electrode spacing is to be 0.050 ±0.001"

(b) Voltage rise rate is to be 600 ±10 volts per second

(c) Five consecutive readings are to be taken on a single sample, with a 3 minute wait between readings. The average of the 5 readings is then calculated. The individual readings and the average are to be recorded.

A satisfactory one-package instrument for this test is the following:

Model 4507 Oil Testing Hypot, manufactured by
Associated Research, Inc.
Chicago, Ill. 60618

e. Limits of acceptability of resistivity and dielectric breakdown voltage.

The minimum acceptable values of dielectric properties of SF 96-1 samples are:

Resistivity at 65 - 85° F: 1.0×10^{10} ohm-cm

Dielectric breakdown voltage, at 68° F minimum (average of 5 readings): 5.0 kilovolts

/s/H.R. BOROSON
H.R. BOROSON
By direction

Copy to: (w/encls)
TRIESTE II,
Electrical/Electronics Officer (2 cys)
NAVSHIPS (SHIPS 2052) (2 cys).

4.7.6 Heating, Vacuum, and Processing

Bacteria and fungi can live and propagate in hydrocarbon fluids. In tropical ocean atmospheres, conditions are optimal for these microorganisms to grow and multiply, especially in oil-water environments. Some opportunities for entry into a compensating system are: in-air fluid exposure, rotating shaft seals, sea-water-exposed check valves, and dynamic O-ring seals (such as compensator piston). Microorganisms and their products can form slime or sludge contaminants in a fluid. This type of contamination will plug filter elements.

An autoclave is often employed to destroy marine organisms. The procedure is to subject a specimen to 259° F steam under 15 psig for 15 minutes. If a fluid were subjected to these elevated temperature conditions in an autoclave, it would be necessary to use a sealed container and allow for volume thermal expansion of the fluid. Another factor is that this high temperature conditioning would probably "thermally age" the fluid and destroy fluid additives. The use of biocide in a fluid appears to be a better approach. However, the feasibility and worth of any reconditioning procedure will be unique for each set of circumstances.

To reiterate, it is desirable to subject a fluid to vacuum to minimize air or gas inclusion. The question must be posed: What level of air inclusion can be tolerated for the particular compensating system? If feasible, gentle stirring of the volume of fluid under vacuum will aid air removal. An increase in temperature will also tend to liberate more dissolved air.

With this in mind, a total fluid reconditioning process will likely involve a combination of several techniques. A loop may be set up where the fluid is continuously filtered, heated, and recirculated. A maintenance procedure may develop so that the fluid is circulated and reconditioned for an hour or more after each mission, every third dive, etc. Heating lowers fluid viscosity, making filtering easier. However, if the subject fluid is vented to atmosphere and the filtering medium will not separate water from the fluid, it will be necessary to actually boil-off water contamination. Low viscosity compensating fluids normally have a low flash point and fire point and are extremely flammable for temperatures which approach the boiling point of water. Hence, the obvious fire hazard for elevated temperature conditioning in the presence of air.

Yet under 28 in. Hg vacuum, it is possible to vaporize water contamination at only 120° F. At a level of 28 in. Hg vacuum and 120° F fluids such as MIL-H-5606B, Skydrol(TM), and Tellus(TM) have vapor pressures such that dissolved air content can be reduced to less than 1% by volume. Water contamination will also "flash off" under these conditions. Conventional vacuum pumps may be used to provide 28 in. Hg vacuum. It is useful to obtain curves which show absolute pressure (into vacuum range) versus saturated air content for various fluids. Select a level of vacuum at a particular temperature which: (1) achieves or exceeds desired air inclusion level (% dissolved air which can be tolerated) and (2) does not exceed the vapor pressure of the fluid. It is possible that vacuum dehydration/degassing will flash off some light ends (move volatile fractions) of low viscosity fluids. The consequent increase in fluid viscosity will likely be negligible, however, if care is exercised on selecting a dehydration temperature and level of vacuum.

Recent studies on water removal from fluids have found vacuum dehydration to be the most effective and desirable method.

Coalescers are reasonably effective, but tend to take additives out of the subject fluid. Although centrifuging will remove gross water contamination from a fluid, this approach is no more effective for removing small concentrations of water.

Commercial conditioning units are also available which degas the fluid at a preselected vacuum level and continuously circulate and filter the fluid to remove water and solids. Salt removal from sea-water contamination of fluids remains a problem area for reconditioning procedures.

4.7.7 Fluid Test Methods

Samples of fluid are ordinarily drained from equipment enclosures during the postdive checkout and are examined to determine water and solids content. One military test reference is NAVSEC Instruction 9210.2 of September 1966, "Submarine Hydraulic Power and Control Systems; Hydraulic Fluid, Cleanliness Requirements for." This instruction is continually examined with respect to its usefulness as an indicator of hydraulic system performance. It is included at the end of this section.

For new construction submarine hydraulic systems, 10-micron absolute or finer filtration is employed.

High cleanliness levels are also required in deep submergence fluid power systems; hence, similar test methods are necessary. While agreement between organizations employing the techniques that are in this instruction sometimes is lacking, the systematic use of these procedures by a single organization on specific equipments will provide a rational basis for assessing fluid and system cleanliness levels. Automatic particle counters are useful and experience with these is being obtained by the Navy.

Techniques such as distillation are somewhat inadequate and/or impractical to measure the amount of sea-water intrusion. In NAVSEC instruction 9210.2, water content is determined by electrometric titration with Karl Fischer reagent. A visual inspection of fluid samples for sea-water contamination is useful, but should not be considered conclusive. For fluids which compensate electrical components and equipments, fluid dielectric properties must be measured. Standard test methods for determining fluid dielectric properties are per ASTM Standards D1169-64 and D877-67. Refer to the silicone fluid reconditioning procedure in section 4.7.5 for a brief description of these procedures.

Applicable Documents: NAVSEC Instruction 9210.2.

C O P Y

NAVSEC INST 9210.2
Ser 6632D3-1477
21 September 1966

NAVSEC INSTRUCTION 9210.2

Subj: Submarine Hydraulic Power and Control Systems; Hydraulic Fluid, Cleanliness Requirements for (Report - NAVSHIPS - 9210-1)

1. Purpose. This Instruction establishes the maximum permissible particulate and water contamination levels for submarine hydraulic systems with operating pressures above 1200 psi. The levels specified are considered to be within the state of the art for initial cleanliness and the capability of installed filtration devices to maintain for reasonable periods of time. The required cleanliness levels are to be obtained prior to delivery, during overhaul, and are to be maintained by the applicable submarines in service.

2. Cancellation. This Instruction cancels and supersedes BUSHIPS INST 9210.17A Ser 632D3-2704 of 21 February 1966.

3. Background. Reliability and life of submarine hydraulic system components can be increased by cleaning-up and maintaining at a high level of cleanliness the hydraulic fluid which is used in the system. Contamination can be present in the form of particulate matter or water. Hydraulic components depend upon the system fluid for lubrication. A contaminated fluid cannot offer the required protection against wear and will generate additional contamination. Water can result in corrosion which causes component maloperation and failure.

The changes to previous cleanliness requirements for submarine hydraulic systems promulgated by this instruction are as follows:

a. Allows the use of a simplified and less time consuming procedure for particle counts by eliminating the counting of particles below 25 microns.

b. Recommends the use of a gravimetrical analysis in addition to the particle count analysis.

c. Modifies the particle count requirements for the various particle size ranges.

d. Establishes an average water content requirement for all the samples from a system in addition to the maximum water content requirement for any sample.

4. Sampling. A minimum of two samples shall be taken for each system located wholly within one compartment. For systems extending into two or more compartments, a third sample shall be taken.

A. Sample points should be marked (labeled) and the same sample points utilized each sampling period. If possible the following sampling locations should be selected:

(1) Location providing sample representative of fluid being supplied to system components.

(2) Return line as close to the supply tank as practical but upstream of any return line filter.

(3) For systems requiring a third sample, a location in the compartment furthest from the pump is recommended.

B. Ideally the sample should be taken from a sampling valve installed specifically for sampling. When sampling valves are not installed the taking of samples from locations where sediment or water can collect, such as dead ends of piping, tank drains, low points of large pipes and filter bowls should be avoided if possible. If sampling from pipe drains allow sufficient flow from the drain before taking the sample to insure a sample more representative of the system. Samples are not to be taken from the tops of reservoirs or other locations where the contamination levels are normally low. If possible, samples are to be taken during system operation following operation of the system in the location from which the sample is taken.

C. Samples may be obtained in bottles cleaned per Fed. Test Method Std. No. 791a, Method 3009.1 (or ARP 598) or by filtering fluid through a grid 0.8 micron filter mounted in a disposable case which is held in a self-cleaning "plug-in" sampler connected directly to the system. When the sampling kit[1] is connected to the system, pressure at the sampling port should not exceed 100 psi during the sampling cycle. On high pressure systems appropriate pressure reducing equipment should be mounted upstream of the sampling port.

5. Sampling frequency: Samples for new construction and in service submarines shall be taken as indicated below:

New Construction Submarines	Submarines In Service
(a) Just prior to first operation of the system	(a) Before overhaul commences
(b) Prior to first sea trial	(b) Prior to first sea trial after overhaul
(c) Just prior to delivery	(c) At completion of overhaul if more than 3 weeks since (b) or if system has been breached
	(d) At approximately 3 month intervals

Ships or activities without analysis capabilities shall submit samples to nearest naval activity with facilities. In accordance with paragraph 082209 of NAVCOMPT Manual Volume 8 the proper requisition form is DD 1146.

6. Water Content Determination Method. Water content shall be determined by ASTM.D1744; Fed. Test Method Std. No. 791a, Method No. 3253, Water by Electrometric Titration with Karl Fischer Reagent. If the water content is suspected to be excessive (in excess of 0.5%), determination of water content can be made by ASTM.D95; Fed. Test Method Std. No. 791a, Method No. 3001.7, Water by Distillation.

7. Particulate Count Method. Particulate counts shall be determined by a procedure based on either Federal Test Method Standard No. 791a, Method 3009.1, "Determination of Solid-Particle Contamination in Hydraulic Fluids" or ARP598 of 3-1-60 issued by the SAE and entitled, "Procedures for the Determination of Particulate Contamination of Hydraulic Fluids by the Particle Count Method" except that the techniques for rapid determination specified herein shall be utilized.

A. Scope. This procedure outlines a rapid method for the determination of particulate contamination in fluid in which a "plug-in" sampling method may be used.

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[1] Available source: Millipore Filter Corporation
Bedford, Massachusetts
Bomb Sampling Kit #XX64 037 00

Particles 25 microns and larger are counted using a rapid microscope scan technique requiring approximately 10 minutes. A two to one (± 33%) maximum variance in results of replicate samples may be expected, provided that the procedure is properly followed.

B. Outline of Method. The fluid is filtered through a white grid (green or black filters may be employed where color contrast is improved for microscopic counting) 0.8 micron filter. A "plug-in" sampler may be connected to the system and system pressure utilized to drive fluid first to flush clean the sampling valve and connections and subsequently through the filter. If a "plug-in" sampling kit is not available the sample may be obtained in a bottle and run through the filter in the laboratory. The contaminants are retained on the filter surface and counted microscopically using oblique incident light with a rapid scan, gating technique. Particles are measured and counted at a single magnification in size ranges above 25 microns only.

C. Microscope Analysis Procedure. A microscope with a mechanical stage, capable of magnification 40-50X is required. Recommended objective is 4-5X with 10X eyepiece. The optimum equipment is a binocular microscope with a micrometer stage. Using the stage micrometer, calibrate the measuring eyepiece. Repeat calibration only as a check or when a new optical component is added. Place filter on the microscope stage and adjust light intensity and position to obtain maximum particle definition. If particles in the 25-100 micron range are estimated to be less than 500, count this range concurrently with the over 100 micron range. If the estimated count exceeds 500 count 10 randomly chosen grid squares from the edge to the center and multiply by ten to obtain the total count.

D. Repeatability. The widest variable in results obtained is caused by the variance between operators in microscope techniques. It is therefore important, in order to establish meaningful and comparable analytical results, that the optical techniques described be followed closely, and that these techniques be checked by means of check samples. Calibration of the ocular micrometer and individual interpretation of the size range limits defined by the ocular micrometer rulings should also be checked by means of calibrated standards. By employing blank analyses for checking the filtration and mounting techniques, and samples for checking the microscope counting aspect of the procedure, facilities may provide themselves with a means for checking the repeatability and reproducibility of the procedure. By analyzing replicate samples from the same sampling port, the facilities may determine the accuracy and reproducibility of sampling procedures.

E. Check Samples. Sample filter discs, permanently mounted and containing representative contaminant from hydraulic fluid systems should be used to check the optical portion of the analytical procedure. The limits of statistical accuracy and, if possible, should be verified by more than one facility. The precision of results obtained should be initially checked and periodically retested by analyzing replicate blanks, samples, and viewing fields in accordance with the procedure. The precision of results obtained in a single facility should also be checked by having all operators concerned occasionally analyze duplicate test samples and compare results.

8. Gravimetric Analysis. An analysis based on Aerospace Recommended Practice ARP 785 "Procedure for the Determination of Particulate Contamination in Hydraulic Fluids by the Control Filter Gravimetric Procedure" is recommended for use by those activities with the equipment necessary for the analysis. The data obtained will be used to evaluate this method of analysis and to establish acceptable cleanliness standards. Initially a maximum particulate contamination of 8mg /100 ml will

be allowed. This analysis is to be used in addition to the particle count analysis and not in lieu thereof except for activities which are not yet equipped for particle counting. The gravimetric method is particularly useful for monitoring flushing operations or analyzing grossly contaminated samples which are difficult to analyze with the particle count method.

9. Contamination Levels.

A. Water Contamination. A water content less than 0.01% is desirable and the water content shall not exceed 0.05% by volume for any sample. The average of samples for any system shall not exceed 0.03%.

B. Particulate Contamination. For new construction submarine hydraulic systems and for submarine hydraulic systems equipped with 10 micron absolute or finer filtration, the particle count for any sample shall not exceed Level A for any size range. For other submarine hydraulic systems on which improved 10 micron absolute filtration has not been installed, the particle counts for any sample shall not exceed Level B for any size range. ShipAlts SS-1006, SSN535 and SSBN-443 provide improved 10 micron absolute filtration for the main and vital hydraulic systems.

Particle Size Range (Microns)	Level A (No. of Particles Per 100 ml.)	Level B (No. of Particles Per 100 ml.)
25-100	7500	30,000
100-500	100	400
Over 500	8	32

No particles over 500 microns except fibers are permitted. Fibers are defined as particles with a length at least ten times the diameter.

10. Information Feedback. Results of samples shall be forwarded to the Naval Ship Engineering Center on NAVSHIPS 5050, NAVSHIPS Report 9210-1. This feedback information shall include the following:

(a) Ship and System Identification

(b) Location from which samples were drawn (sketch or plan reference with first report)

(c) Water content and particle count contamination levels.

(d) Activity performing analysis.

(e) System status at time of sampling as described in paragraph 5 plus time system is operated in location of sampling valve immediately prior to sampling.

(f) Any unusual presence of undesirable wear products such as brass, bronze or other metals should be noted under remarks. Explanation for these products if known, such as pump failure or source of sea water contamination should be noted.

11. System Cleanliness Requirements. Where samples exceed the limits established by this Instruction, the system shall be cleaned by filtering, draining and refilling, or flushing as required to meet the acceptable cleanliness levels. Prior to cleaning, resampling to verify analysis is recommended.

12. Reporting. NAVSHIPS report symbol 9210-1 has been assigned to this report requirement. NAVSHIPS 5050 is available from Cognizance I symbol stocking points (NORVA and OAKLAND).

CHAPTER V

COMPENSATING SYSTEM SURVEYS

5.1 INTRODUCTION

Throughout this investigation, current and past literature sources, articles, and technical reports have been scrutinized for information. In addition, survey sheets were mailed to 50 civilian and Navy organizations as a means to retrieve design and operational information for fluid-filled, depth/pressure-compensated systems, such as:

- Electrical distribution systems.
- Hydraulic systems.
- Power conversion and control components.
- Electric motors.
- Speed reducers.

Survey contributors have validated what is the realistic "state of the art" in compensating systems design. The cooperation received has aided this effort materially. These organizations are acknowledged in table 5.2.

5.2 TABLE OF CONTRIBUTING ORGANIZATIONS

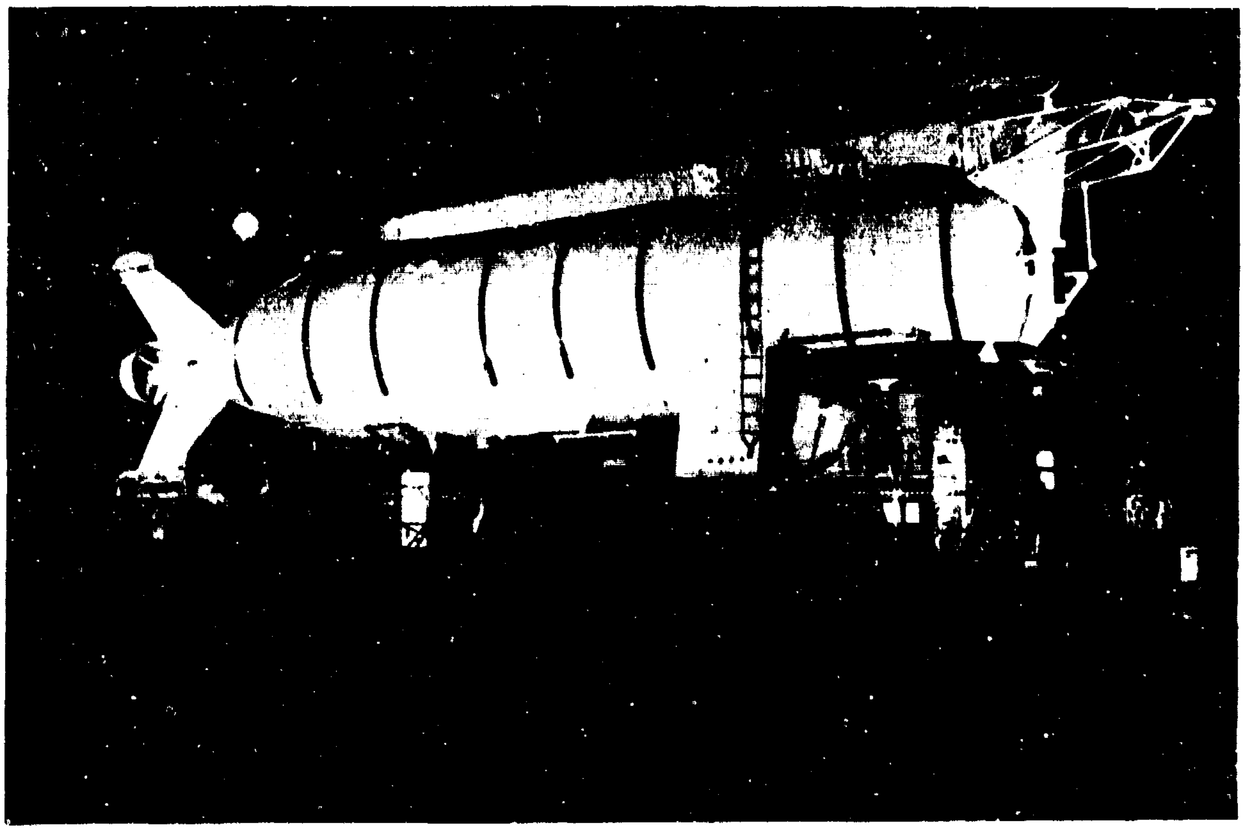
Organization	Individual
B.F. Goodrich Co. Akron, Ohio	W.E. Schlag
Benthos, Inc. North Falmouth, Mass.	S.O. Raymond
Edrede Corp. Colorado Springs, Col.	D.L. Crabtree
Franklin Electric Co. Bluffton, Ind.	S.G. Makowski R.S. Nowakowski
General Dynamics Corp. Electric Boat Div. Groton, Conn.	V.T. Boatwright, Jr. T.J. Gerken M.A. Gruss G. Lefoley
Interstate Electronics Corp. Oceanics Div. Anaheim, Calif.	R.H. Jones
Kobe, Inc. Huntington Park, Calif.	P.H. Jamison
Lockheed Missiles & Space Co. Ocean Systems Engineering Sunnyvale, Calif.	E.P. Pallange
Mare Island Naval Shipyard Vallejo, Calif.	J. Clerici
Naval Civil Engineering Laboratory Port Hueneme, Calif.	S.L. Bugg G.L. Liffick P.K. Rockwell S. Black R. Brackett
Naval Ship Research and Development Laboratory Panama City, Fla.	W.B. Culpepper R.L. Bentz
Ocean Systems, Inc. Reston, Va.	B.C. Gilman R.P. Bishop R.J. Dzikowski
Pesco Products Bedford, Ohio	C.W. Hughes F.G. Johnston
Reda Pump Co. Bartlesville, Okl.	J.C. O'Rourke
Scripps Institution of Oceanography San Diego, Calif.	D.K. Gibson O. Kirsten
Singer- General Precision, Inc. Kearfott Division Clifton, N.J.	A.H. LeFebvre
Southwest Research Institute San Antonio, Texas	E.M. Briggs
Underseas Engineering, Inc. Riviera Beach, Fla.	A.G. Anderson
Westinghouse Electric Corp. Ocean Research and Engineering Center Annapolis, Md.	R.W. Peach C.B. Barclay, Jr.

5.3 SELECTED COMPENSATING SYSTEM SURVEYS

It was not feasible to publish the bulk of survey material and additional information which was received by this activity. However, several thorough and informative surveys have been included to stimulate a general train of thought for this design application.

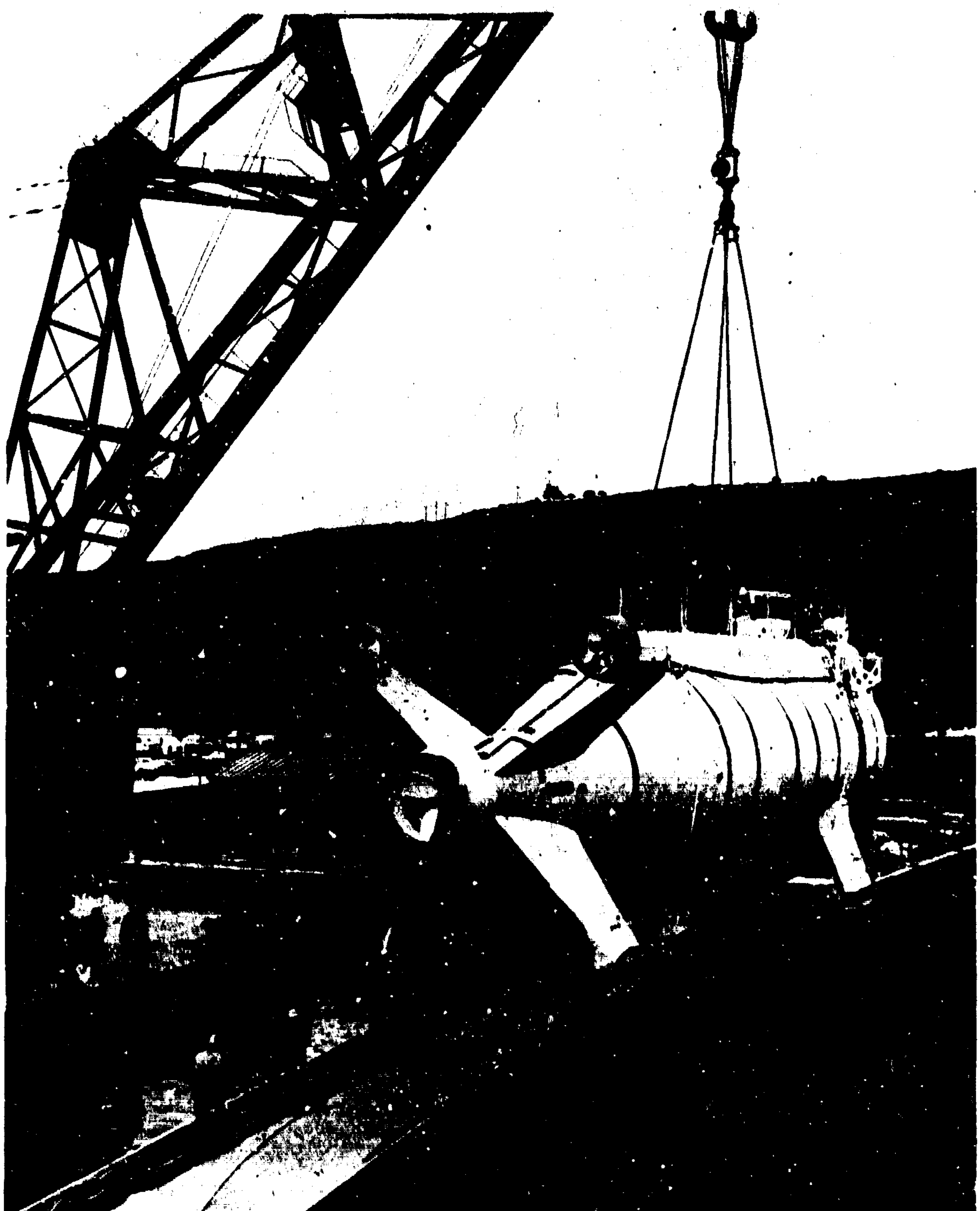
The information contained in these surveys is presented with the originator's approval and its presentation does not indicate its adequacy by the Department of the Navy.

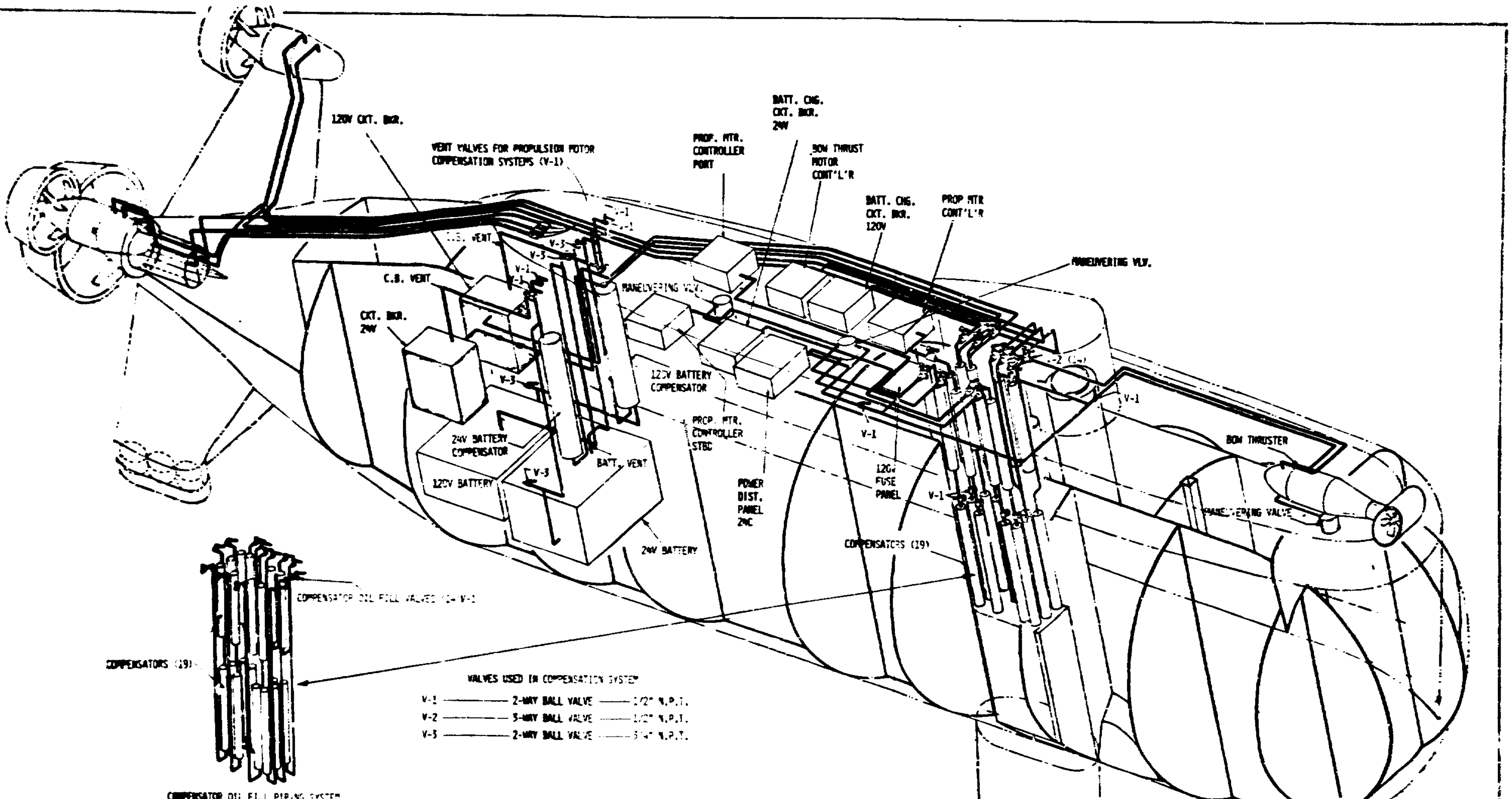
- TRIESTE II Bathyscaphe
- AUTEC I & II
- DEEPSTAR 20000
- DSRV-I



TRIESTE II Bathyscaphe

5.3.1 TRIESTE II Bathyscaphe





TRIESTE II

ELECTRICAL COMPENSATION SYSTEM

NOTE:
VENTING FOR THE MANEUVERING VALVES, BOW THRUSTER, PROPULSION MOTOR CONTROLLERS, BOW THRUSTER MOTOR CONTROLLER, BATTERY CHARGING CIRCUIT BREAKERS, 24V POWER DISTRIBUTION PANEL AND 120V FUSE PANEL IS PROVIDED FOR BY V-1 VALVES (NOT SHOWN) LOCATED AT EQUIPMENT.

Fluid Pressure-Compensating Systems
Survey Information Sheet

Date: 1-15-71

Contributing Organization, address:
Mare Island Naval Shipyard
Ocean Engineering, Code 280
Vallejo, Calif. 94592

Name, address, phone no. of contributor(s):

J. Clerici Code 283J (707)646-4183

I. Submersible name, or submersible system:

TRIESTE II

Designer (firm or organization):

Ocean Engineering, Code 280

Builder:

Mare Island Naval Shipyard

Design depth: **20,000 ft**

Test depth: **30,000 ft**

Operational date:

Description of fluid pressure-compensated system, general operational requirements of system:

Systems include propulsion motor, circuit breaker, control and fuse panels. Each system is compensated by a vertical cylindrical rubber tube located at a point physically lower than the device to be compensated; thereby providing a positive pressure for each compensated enclosure. The pressure is relative to the head pressure created by the compensator fluid and sea water specific gravity difference.

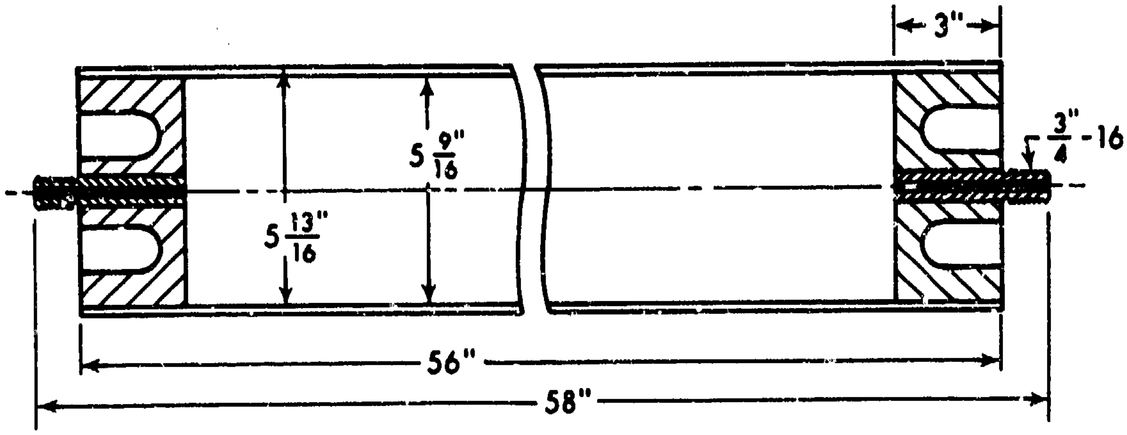
II. Compensating device

A. Configuration (brief description):

Rubber tube approx 5 13/16" X 56" in length standing vertical in a tank open to sea pressure. Top of tube is connected via a flexible tube to the system, the bottom end is brought out to a fill or vent valve.

1. Special, or modification of existing hardware; supplier:

2. Drawing or sketch of device:



3. Ambient temperature range (design); what environmental considerations these temperature extremes represent:

35° - 120° F service range.

sea bottom to surface

4. Average operating temperature of fluid-filled system:

per range

5. Ambient pressure range of operation:

0-10000 psi

6. Overpressure or positive bias over ambient sea-water pressure, Δp or unpressurized:

< 2.5 psi positive.

Rationale for pressurized versus unpressurized or the converse:

Leak out in lieu of leak in.

7. Design rationale for selection of this configuration of compensator as related to system requirements:

Selected on basis of extreme weight savings, positive Δp with no mechanical springs, low cost, good longevity, space utilization, no corrosion, extremely small dead band (inactive fluid).

8. Probable failure mode. Any redundancy to compensation, provision to be sea-water floodable as worst condition? Will system "fail-sick" or "fail-dead"?

   1. **No service failures to comp systems to date. Extreme fluid losses caused by enclosure stuffing tube failures caused total depletion of compensator with no damage to compensator tube. Flooding of compensated device was ultimate failure.**
   2. **One tube failure caused by overfilling pressure, tube burst.**
   3. **Tubes have shown chaffing wear from proximity to interferences in tank.**

B. Materials selection for compensated system

Compensator housing or case material. Bare metal exterior or protective coating, paint? **Rubber**

Refer to manufacturing rubber procedure

enclosure one.

Heat transfer considerations, if any:

Elastomer type, physical properties, any MIL-SPEC? Supplier:

enclosure one. (Rubber Lab Inst 113)

If spring is used, what type, material, and is it exposed to sea water?

none

Use of external screens, baffles, or filters to minimize damage from sea-water particle contamination on elastomer or piston and internal housing. Give details, mesh size, or filtration:

none

Marine fouling, corrosion protection:

none

(a) Paints and coatings, metal passivation

none

(b) Galvanic protection

none

(c) Use of protective greases, lubricants, bedding compounds

none

Estimated safety factor, materials life:

Materials life est. 10 years

III. Compensating fluid considerations

A. Fluid selection

1. Type or trade name:

   Dow Corning DC-200-1 silicone

2. Properties of fluid, as related to operation of system:

   1. **Low viscosity for d-c propulsion motor; electrical relay and contact applications.**
   2. **Low specific gravity 0.818±0.004 at 25° C/25° C. provides positive system pressure.**

3. Emulsifying or nonemulsifying; rationale for this decision:

   Not applicable

4. Materials compatibility problems for compensated system? If so, cite problems and remedy.

   1. **All materials checked for fluid compatibility prior to use. Unsuitable material deleted.**
   2. **Poor lubricity for (1) cs silicone fluid created propulsion motor bearing and comutator problems. Overcome by changing from steel to plastic ball bearing ball retainers and addition of lubricating additive, dioctyl tetrachlorcpthalate 5 % by volume.**

B. Design Parameters for sizing of compensator
Please show sample calculations of design scheme on extra sheet.
Enclosure Volume (2) and Temperature Calculations

Compensating fluid type **silicone** Viscosity **(1) cs**
Overall temp range **35-120 service** ° F.
Overall pressure range **0-10000** psig.

Was this compensator designed for a specific environmental range of operation or for the extremes of arctic and tropical conditions? Why?

1. Compressibility factor, at depth ______ (bulk modulus effect).
   Est. Volume change ______ %
2. Equipment storage, transportation
   Pressure ______
   Temp. range, ° F ______
   Est. volume change, heating ______ %
   Cooling ______ %
3. Operational Characteristics
   Fluid coefficient of thermal expansion ______ /° F

See Enclosure (3) Typical comp sys sketches

   Surface Conditions
   Max. operating temp.,° F **120°**
   Max. ambient sea-water temp., ° F **85°**
   Est. volume change ______
   Depth Conditions
   Max. operating temp., ° F ______
   Lowest ambient sea-water temp., ° F **35°**
   Est. volume change ______
4. Seal Leakage (est.) **unknown** cc/hr
   Volume change for length of mission ______ %
   Avg. shaft rpm or reciprocating rate **150 rpm**
   Type of seal, where used **crane shaft seal**
5. Sea-water intrusion (est.) **unknown** cc/hr
   Where may it enter system?
   Volume change for length of mission ______ %
6. Other ______

   Total usable compensator volume expressed as % of fluid volume in compensated system **>95** % **Enclosure (4) (tube diff press. curves)**

   Has this volume proved to be adequate for your system? Comment **yes**

   Total fluid volume in compensated system **1215 in.^{3} per comp tube**
   19 tubes in all.

IV. Special design requirements

A. Use of protective devices in compensated system. Enclose system diagram, if possible.

1. Sea-water leak detectors. Description, location.

none

2. Vent or relief valves

(a) Fluid expansion relief
Valve configuration, location, relief pressure:

No relief valves. Compensated system filled to capacity and measured amount extracted for expansion. Safety margin provided in elastic quality of tube.

(b) Electrical equipment gassing relief
Valve configuration, location, relief pressure:

(c) Valve discharge port protection from intrusion of sea water?
Sketch

3. Fluid volume indicators or device to monitor compensator volume:

4. Temperature monitoring. Give type of sensor and location.

5. Differential pressure transducer (for Δp between compensating fluid and ambient sea-water pressure).

6. Other

B. Fluid circulation devices in rotating machinery to aid in cooling the compensating fluid. Any use of fluid filtering and/or cooling devices for electric motors, speed reducers, hydraulic pumps or motors, etc.

C. Sea water to compensating fluid seals in system

1. Type, MIL-SPEC No. (if known), material, where used, estimated leakage rate.

   (a) Static seals

Ethylene propylene "O" rings or viton	**MIL-S-68851 70 DUR EP-62**

   (b) Rotating or reciprocating seals

   Crane shaft seals w/stellite
   "O" ring floating seal J. Crane Ty. 1
   Bul S213-3 Fig A w/Viton bellows.

   (c) Seal problems, remedy -

   Rotating seals required added lapping for initial installation to control static condition leaks.

2. Use of protective devices for outboard shaft seals, including actuators:

D. Fluid ports or penetrations, tube fittings, quick-disconnects

1. Give type, material, use in compensating system.

   CL 316 Stainless, Monel and Bronze

2. For quick-disconnects, what type of protective caps or plugs are used to prevent sea-water intrusion?

E. Other unique, desirable design features

All fill and vent valves to compensated devices located in wet (surfaced) areas located to permit easy access.

V. Compensating system operation, maintenance

A. Physical system arrangement. Show diagram, identifying components.

Enclosure (5) (Typical ARRGT sketch)

B. System fill, vent procedure

Enclosure (6) (System OPS sketch)

C. Predive positioning of compensator or final volume adjustment.

Enclosure (6)

D. Predive and postdive compensating system checkout. Explain contamination removal, refilling, and venting procedure.

Enclosure (6)

E. Fluid change interval
Renewal dictated by normal maint to equip. No set fluid change sched.

F. Drain and flush procedures

1. Drain elect enclosure via sumps flush with fresh fluid.
2. Back flush comp tubes via valve systems.

G. Fluid reconditioning procesc, if any

Manufacturers reclaim

H. For compensator overhaul, describe cleaning and reconditioning of metal and elastomeric parts - what cleaning solutions or solvents are used?

Discard and replace

I. Compensator test methods, if any

Air test to 10 psi, 40 psi proto-type.

VI. Applicable Documents

Manuals, MIL-SPECS, design standards, etc that were used as a reference in the design and fabrication of the compensating system.

Fluid MIL-S-21568A

PROCEDURE FOR MANUFACTURING COMPENSATING TUBE FOR TRIESTE II BATHYSCAPH SUPPORT

Mare Island Rubber Laboratory Instruction No. 113

1. SCOPE

1.1 This instruction describes the procedure for manufacturing compensating tubes for TRIESTE II as shown in Mare Island Sketch 270-516-004A. These are neoprene (rubber) tubes with fabric reinforcement. Each tube is 5-9/16 inches inside diameter by 56 inches long. Both ends of the tube are closed by means of rubber plugs. A metal insert, CRES 316, extends through each rubber plug and is bonded thereto.

2. SAFETY PRECAUTIONS

2.1 The Hylene M-50 which is used to treat the fabric is an isocyanate. The vapors from the Hylene M-50 in toluene may cause allergic reactions in the respiratory system. Adequate ventilation must be provided in the area, during the drying of the treated fabric. If allergic reactions occur, ventilation must be increased and workers must wear organic vapor respirators during the period of fabric dipping and drying. Polyethylene or rubber gloves shall be worn to protect the skin.

2.2 The rubber cement and the toluene used in other operations are flammable. Adequate ventilation shall be provided and free flames shall not be permitted.

3. MATERIALS

3.1 Spun nylon fabric Style SN-109, 4.10 ounces per square yard, 40 inch width, 44 by 44 count. Manufactured by Wellington Sears Co., Inc.

3.2 Hylene M-50. Manufactured by E.I. du Pont de Nemours Co., Inc.

3.3 Toluene. Stock No. 6810-290-0046 in 55-gallon drum.

3.4 Stock N-64-1.

3.5 Chemlok 205 and Chemlok 220. Manufactured by Hughson Chemical Co.

Pressure vs Volume

V_s - sum of the volume of oil needed to fill unit under pressure before the unit is pressurized

V_o - volume of oil required to fill unit to be compensated

$$V_s = V_o + V_o r + V_o r^2 + V_o r^3 + ---- + \ + V_c r^n$$

$$V_s = V_o \sum_{n=0}^{\infty} r^n$$

$$V_s \xrightarrow[\lim\ n \to \infty]{} \left(\frac{1}{1 - r}\right) V_o \quad \text{when } |r| < 1$$

For silicone oil

$r = .0485$ at 10,000 psi & 32° F

$$\therefore V_s = V_o\left(\frac{1}{1 - .0985}\right) = 1.05097\ V_o$$

For silicone oil

$r = .072$ at 10,000 psi & 70° F

$$V_s = V_o\left(\frac{1}{1 - .072}\right) = 1.0776\ V_o$$

Temperature vs Volume

V_t - volume needed due to thermal expansion

For silicone oil thermal expansion is .00130 cc/cc/° C

= 5/9(.00130) = .000722 unit/unit/° F

$$\therefore \alpha = .072\% \text{ reduction/° F}$$

$$\therefore V_t = V_s + V_s \alpha \Delta T$$

$$V_t = V_s(1 + .00072\Delta T)$$

Assume $T_{max} = 120°$ F

At 32° F

$$V_s = 1.05097\ V_o$$

$$V_t \approx (1.051)V_o[1 + .000722(120 - 32)]$$

$$= (1.051)(1 + .063586)V_o = (1.051)(1.063586)V_o = 1.1187\ V_o$$

The amount of increase in volume is $1.119\ V_o - V_o = 11.876\%$. This is 11.9% increase in volume.

At 70° F

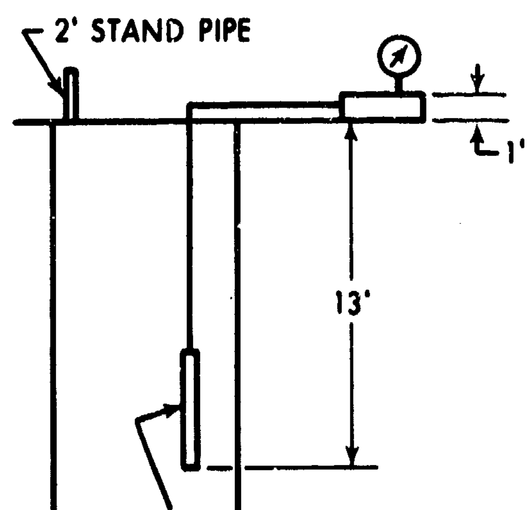
$$V_s = 1.0776\ V_o$$

$$V_t = (1.0776)V_o[1 + .000722(120 - 70)] = 1.0776\ V_o(1 + .0361)$$

$$= 1.0776(1.0361)V_o = 1.1165\ V_o \approx 1.117$$

Change in volume = $1.117\ V_o - V_o = .117\ V_o \Longrightarrow$ 11.7% increase in pressure on electrical boxes.

CASE I - Box on Deck and Compensator on Lower Level



SUBMERGED

$$\Delta P_1 = .09(10) = .9 \text{ psi at } 20{,}000 \text{ ft}$$

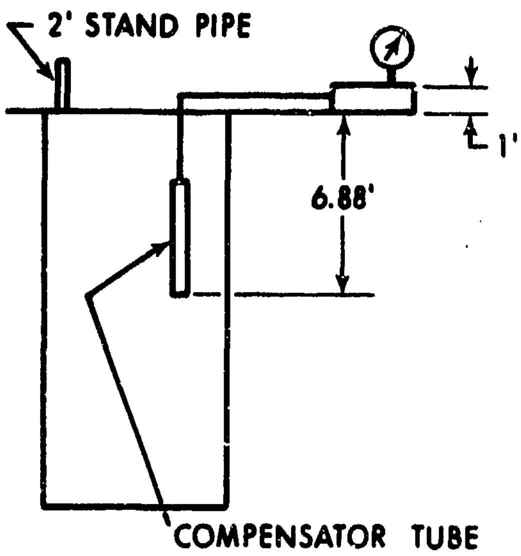
$$\Delta P_2 = (.09)(14) = 1.26 \text{ psi at } 0 \text{ ft}$$

SURFACED

$$P = .444(15) - .354(14)$$

$$= 6.65 - 4.95 = 1.7 \text{ psi}$$

CASE II - Box on Deck and Compensator on Upper Level



SUBMERGED

$$\Delta P_1 = (.09)(2.88) = .259 \text{ psi at } 20{,}000 \text{ ft}$$

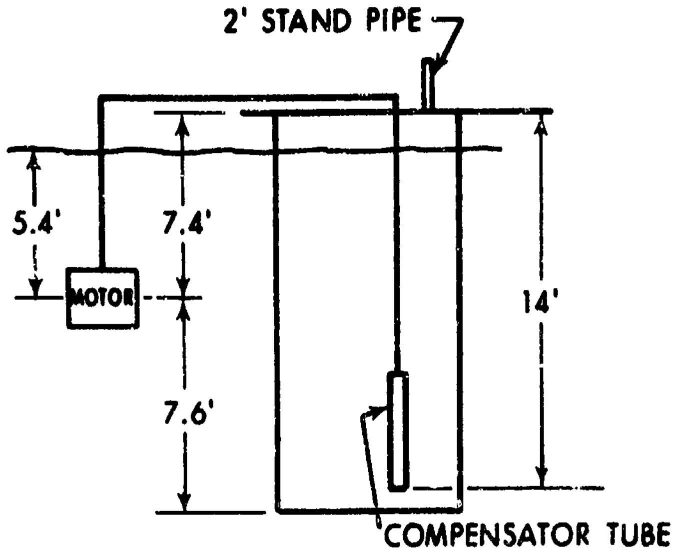
$$\Delta P_2 = (.09)(6.88) = .618 \text{ psi at } 0 \text{ ft}$$

SURFACED

$$P = (8.88)(.444) - (7.88)(.354)$$

$$= 3.95 - 2.79 = 1.16 \text{ psi}$$

CASE III - ℄ Motor and Compensator



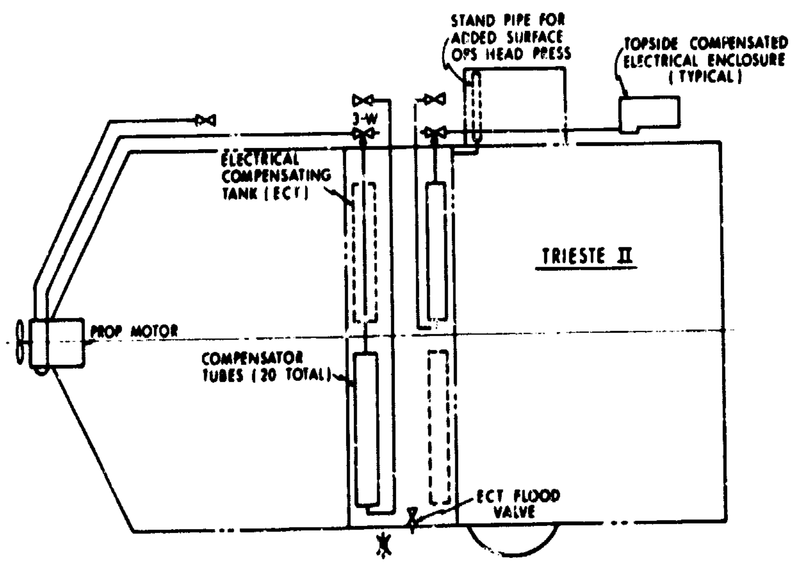
SUBMERGED

$$\Delta P_1 = .09(10 - 7.4) = .09(2.6) = .234 \text{ psi at } 20{,}000 \text{ ft}$$

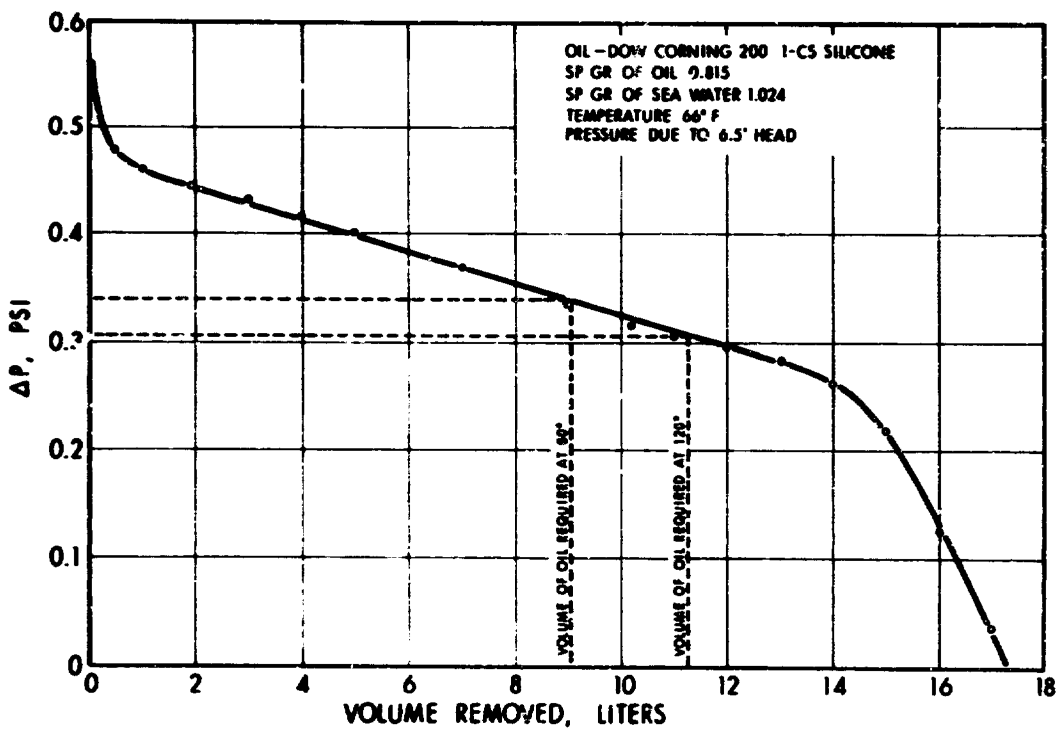
$$\Delta P_2 = .09(14 - 7.4) = .09(6.6) = .594 \text{ psi at } 0 \text{ ft}$$

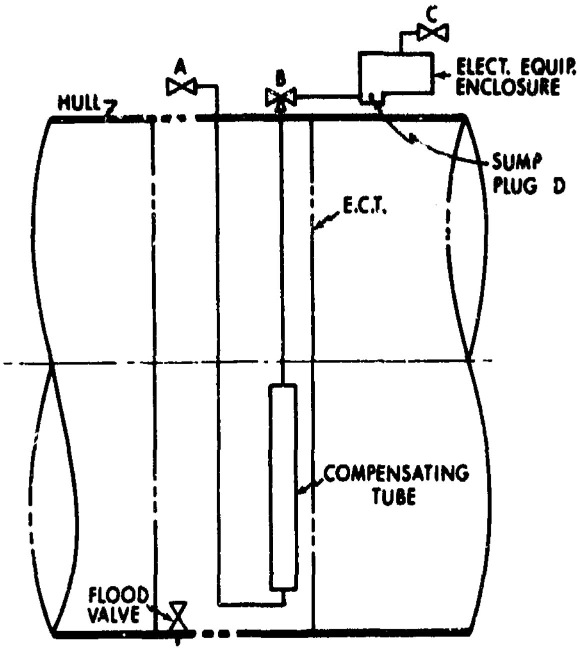
SURFACED

$$\begin{aligned} P &= 16(.444) - 14(.354) + 7.4(.354) - 5.4(.444) \\ &= 10.6(.444) - 6.6(.354) \\ &= 4.71 - 2.33 \\ &= 2.38 \text{ psi} \end{aligned}$$



Compensator Differential Pressure



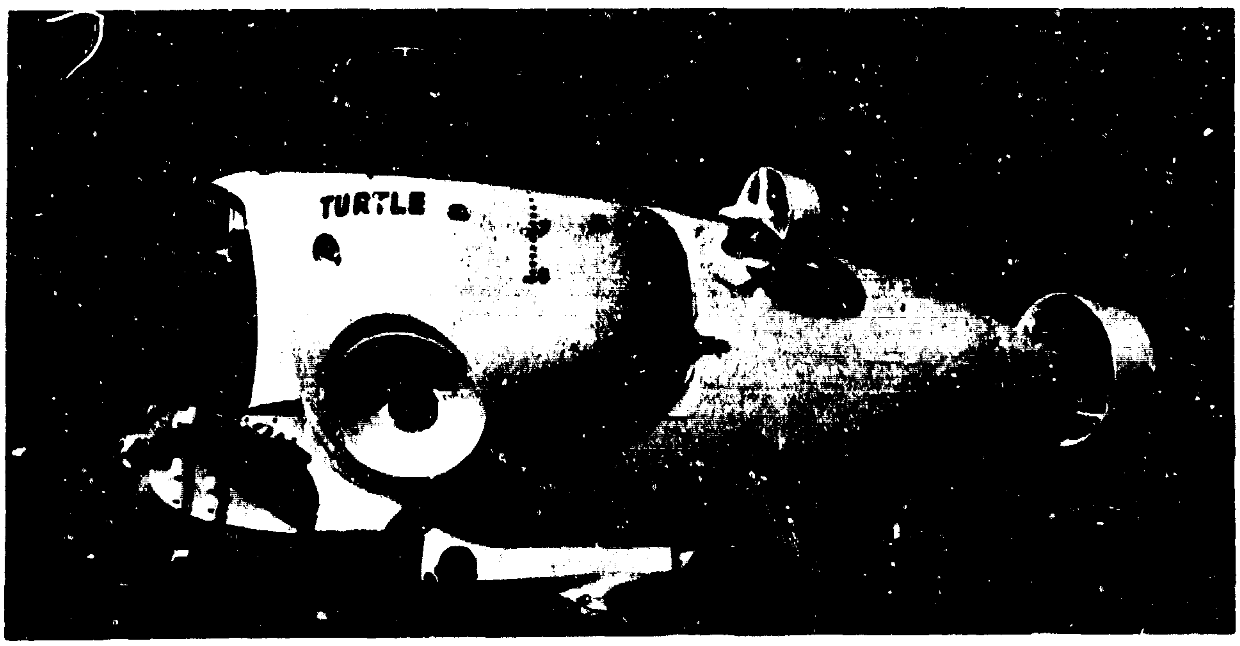


System Operation			
Valve	**Operation**	**Valve Position**	**Remarks**
A	Initial fill	O	Fluid supply connected
B		↱	To equip.
C		O	Vent
A	Temp adjust and/or pre-dive	O Limited	Air pressure or water in ECT
B		↱	To equip.
C		O	Supply pump attached
A	Dive	X	
B		↱	To equip.
C		X	
A	Leak det	X	
B		↰	To vent
C		O	Fluid supply connected
D		O	As req'd

X = closed
O = open
↰ = flow direction

TRIESTE II Typical Comp System ~ Operation

5.3.2 AUTEC I and II



Fluid Pressure-Compensating Systems
Survey Information Sheet

Date: 1-29-71

Contributing Organization, address:
General Dynamics Corporation
Electric Boat Division
Groton, Connecticut

Name, address, phone no. of contributor(s):

T.J. Gerken 203-446-6272

I. Submersible name, or submersible system:

AUTEC I & II (Sea Cliff and Turtle)

Designer (firm or organization):
General Dynamics Corporation
Electric Boat Division

Builder:
General Dynamics Corporation
Electric Boat Division

Design depth:
6500 ft

Test depth:
6500 ft whole vehicle
110 % pressure hull
150 % most outbd components

Operational date:
Sept. 1970

Description of fluid pressure-compensated system, general operational requirements of system: HYDRAULIC SYSTEM, pp. V-25 to V-35

1-One enclosure containing hydraulic pumps, solenoid valves, and instrumentation transducers. Several hydraulically driven auxiliaries are compensated by their return lines to this enclosure.
2-Two propulsion reduction gears and one propulsion training gear.
3-One enclosure containing hydraulic motor-pump and solenoid valves (trim system).
4-One enclosure containing hydraulic motor-pump, solenoid valve, and instrumentation transducer (variable ballast system).
5-One enclosure containing solenoid valves and instrumentation transducer (water ballast system).

II. Compensating device

A. Configuration (brief description):

For units 1 & 2, piston-cylinder type with spring and diaphragm seal

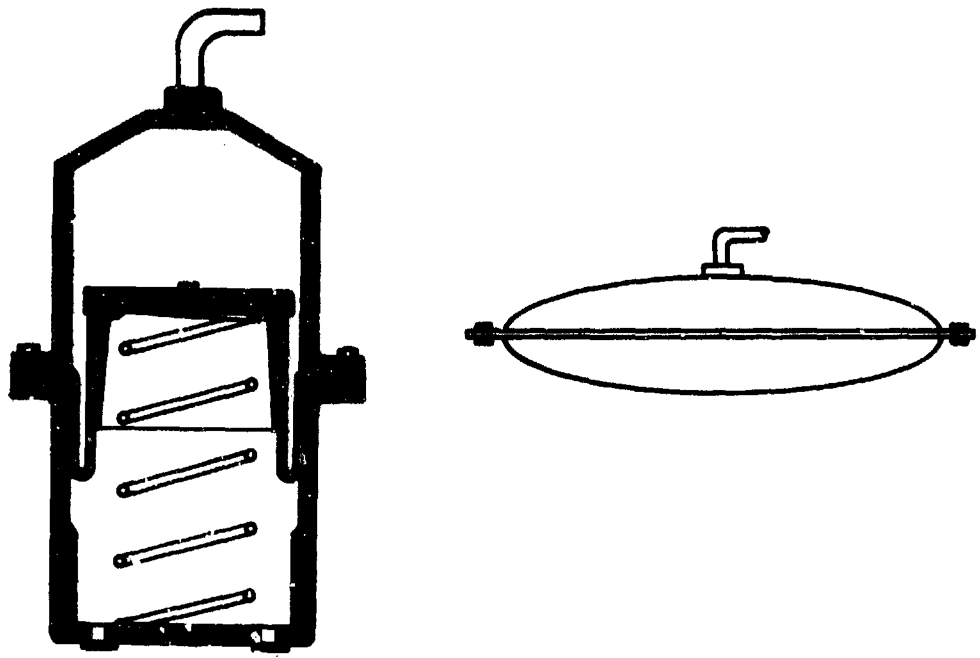
For units 3, 4, & 5 slack diaphragm type

1. Special, or modification of existing hardware; supplier:

Designed for specific application by GD/EB

Purchased from B.F. Goodrich to their specific design

2. Drawing or sketch of device:



3. Ambient temperature range (design); what environmental considerations these temperature extremes represent:

0 - 130° F	**Tenderborne air ambient**
27 - 95° F	**Waterborne ambient**

4. Average operating temperature of fluid-filled system:

   55°

5. Ambient pressure range of operation:

   0 - 3000 psig

6. Overpressure or positive bias over ambient sea-water pressure, Δp or unpressurized:

2 - 4 psi	**0 psi**

   Rationale for pressurized versus unpressurized or the converse:

Biased when moving shaft seals are present in compensated device	**Not biased when moving shaft seals are not present in compensated device**

7. Design rationale for selection of this configuration of compensator as related to system requirements:

Perform function with minimum weight

8. Probable failure mode. Any redundancy to compensation, provision to be sea-water floodable as worst condition? Will system "fail-sick" or "fail-dead"?

- **Outward leakage of fluid until compensator is exhausted, then exchange of fluid and sea water**	- **Exchange of fluid and sea water**
- **No redundancy**	- **same**
- **Hydraulic equipment can continue to operate for short time with water present**	- **same**
- **Electrical connections near bottom of enclosure waterproofed for continued short time operation in presence of limited flooding**	- **same**

B. Materials selection for compensated system

Compensator housing or case material. Bare metal exterior or protective coating, paint?

Monel, Bare **Rubber**

Heat transfer considerations, if any:

Temperature rises modest; no special provisions waterborne

Elastomer type, physical properties, any MIL-SPEC? Supplier:

Buna N, no Mil Spec

If spring is used, what type, material, and is it exposed to sea water?

Be Cu, yes

Use of external screens, baffles, or filters to minimize damage from sea-water particle contamination on elastomer or piston and internal housing. Give details, mesh size, or filtration:

NiCu, 10 x 10 Mesh

Marine fouling, corrosion protection:

Marine growth not present since vehicles are only waterborne when actually in use.

(a) Paints and coatings, metal passivation

Devran paint on aluminum compensated enclosures.

(b) Galvanic protection

Proper choice of materials; no cathodic protection system present.

(c) Use of protective greases, lubricants, bedding compounds

none

Estimated safety factor, materials life:

III. Compensating fluid considerations

A. Fluid selection

1. Type or trade name:

   Hydraulic fluid, MIL-H-5606B

2. Properties of fluid, as related to operation of system:

   - **Customary properties for hydraulic use.**
   - **Electrically non-conducting.**
   - **Void filler.**
   - **Lubricity for gears.**

3. Emulsifying or nonemulsifying; rationale for this decision:

   Fluid and water will emulsify upon rapid churning, but this is not a desired property. Preferable that minor intrusion of water remain separated in bottom of system.

4. Materials compatibility problems for compensated system? If so, cite problems and remedy.

   Elastomers must be selected to be compatible with fluid and sea water, but suitable materials are readily available.

B. Design Parameters for sizing of compensator

Please show sample calculations of design scheme on extra sheet.

Compensating fluid type Hydraulic oil, MIL-H-5606B Viscosity ________

Overall temp range 27-95 Waterborne ° F. 0-130 Tenderborne

Overall pressure range 0-4500 psig.

Was this compensator designed for a specific environmental range of operation or for the extremes of arctic and tropical conditions? Why?

Specification requirement

1. Compressibility factor, at depth ______ (bulk modulus effect).
   Est. Volume change - 1.6 %
2. Equipment storage, transportation
   Pressure Atmospheric
   Temp. range, ° F 0-130
   Est. volume change, heating ________ %
   Cooling ________ % TTL 5.3 %
3. Operational Characteristics
   Fluid coefficient of thermal expansion 0.00041 /° F

   Surface Conditions
   Max. operating temp.,° F ________
   Max. ambient sea-water temp., ° F ________
   Est. volume change ________
   Depth Conditions
   Max. operating temp., ° F ________
   Lowest ambient sea-water temp., ° F ________
   Est. volume change ________

   For design, max vol occurs at 130 F and 0 psig; min vol occurs at 27 F and 4500 psig. Difference is 4.2 % for temp and 1.6 % for pressure. These changes are additive, thus totaling 5.8 %
4. Seal Leakage (est.) Allow for ~ 20 cc/hr
   Volume change for length of mission ~ 2 %
   Avg. shaft rpm or reciprocating rate 1800 RPM
   Type of seal, where used LIP

   For 100 hrs, by spec; mission is ≈ 8 hrs
5. Sea-water intrusion (est.) None cc/hr
   Where may it enter system?
   Volume change for length of mission ________ %
6. Other ________

Total usable compensator volume expressed as % of fluid volume in compensated system ________ %
varies with individual system, but approx 8 %

Has this volume proved to be adequate for your system? Comment ________

Yes. Extreme environmental conditions used for design are not actually attained in service. See also para. V.C. below.

Total fluid volume in compensated system varies with individual system, over range of 1 to 5 ft^3.

IV. Special design requirements

A. Use of protective devices in compensated system. Enclose system diagram, if possible.

1. Sea-water leak detectors. Description, location.

CRES electrode approximately 1/8 inch above lowest point of enclosure. Alarm at approximately 10^3 ohms from electrode to enclosure. Used only in enclosures containing electrical equipment. Drain or sampling plug generally located close to electrode location.

2. Vent or relief valves

(a) Fluid expansion relief
Valve configuration, location, relief pressure:

2-4 psid, location generally near top of canister but not critical.

(b) Electrical equipment gassing relief
Valve configuration, location, relief pressure:

No normal gassing for these systems.

(c) Valve discharge port protection from intrusion of sea water?
Sketch

None; valve operates in sea water.

3. Fluid volume indicators or device to monitor compensator volume:

Portable pressure gage; visual or tactile methods; all with vehicle tenderborne.

4. Temperature monitoring. Give type of sensor and location.

None.

5. Differential pressure transducer (for Δp between compensating fluid and ambient sea-water pressure).

None.

6. Other

B. Fluid circulation devices in rotating machinery to aid in cooling the compensating fluid. Any use of fluid filtering and/or cooling devices for electric motors, speed reducers, hydraulic pumps or motors, etc.

None necessary waterborne.

C. Sea water to compensating fluid seals in system

1. Type, MIL-SPEC No. (if known), material, where used, estimated leakage rate.

   (a) Static seals

   O-rings, flat gaskets

   (b) Rotating or reciprocating seals

   Lip seals.

   (c) Seal problems, remedy -

2. Use of protective devices for outboard shaft seals, including actuators:

   none

D. Fluid ports or penetrations, tube fittings, quick-disconnects

1. Give type, material, use in compensating system.

   MS type or quick disconnect type for almost all connections. All CRES.

2. For quick-disconnects, what type of protective caps or plugs are used to prevent sea-water intrusion?

   All quick disconnects are mated before vehicle is waterborne.

E. Other unique, desirable design features

V. Compensating system operation, maintenance

A. Physical system arrangement. Show diagram, identifying components.

Compensators are connected to their respective enclosures by pipe, flex hoses, and quick disconnects.

B. System fill, vent procedure

Quick disconnect at a high point is broken, and enclosure and compensator are separately filled and vented.

C. Predive positioning of compensator or final volume adjustment.

While compensators are designed for fluid expansion as well as contraction, as a practical matter they are often topped-off just prior to launching so as to increase available compensating margin. This may require bleeding after retreival in warm operating areas.

D. Predive and postdive compensating system checkout. Explain contamination removal, refilling, and venting procedure.

Pre-dive as in C. above. Post-dive check for fluid loss is made.

E. Fluid change interval

F. Drain and flush procedures

G. Fluid reconditioning process, if any

Not economically attractive.

H. For compensator overhaul, describe cleaning and reconditioning of metal and elastomeric parts - what cleaning solutions or solvents are used?

I. Compensator test methods, if any

VI. Applicable Documents

Manuals, MIL-SPECS, design standards, etc that were used as a reference in the design and fabrication of the compensating system.

Specification for building a research vehicle AUTEC I.

Fluid Pressure-Compensating Systems
Survey Information Sheet

Date: 3-1-71

Contributing Organization, address:
General Dynamics Corporation
Electric Boat Division
Groton, Connecticut

Name, address, phone no. of contributor(s):

T.J. Gerken 203-446-6272
M A. Gruss 203-446-2559

I. Submersible name, or submersible system:

AUTEC I & II (Sea Cliff and Turtle)

Designer (firm or organization):

General Dynamics Corporation
Electric Boat Division

Builder:

General Dynamics Corporation
Electric Boat Division

Design depth:

6500 ft

Test depth:

6500 ft
110 % pressure hull
150 % most outboard component

Operational date:

Description of fluid pressure-compensated system, general operational requirements of system: ELECTRICAL SYSTEM, pp. V-36 to V- 46

1- Five DC electric motors, aggregating 20 HP.

2- Two electric distribution centers, containing principally circuit breakers and motor controllers.

II. Compensating device

A. Configuration (brief description):

For Item 1, piston-cylinder type with spring and diaphragm seal

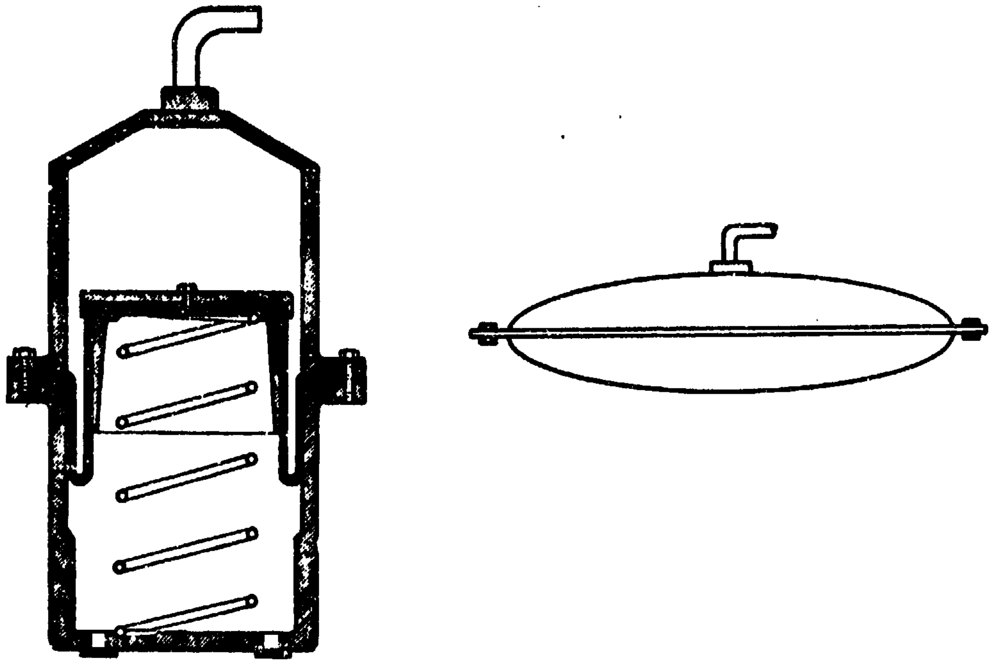
For Item 2, slack diaphragm type

1. Special, or modification of existing hardware; supplier:

Designed for specific application by GD/EB

Purchased from B. F. Goodrich to their specific design

2. Drawing or sketch of device:



3. Ambient temperature range (design); what environmental considerations these temperature extremes represent:

0 - 130° F	Tenderborne air ambient
27 - 95° F	Waterborne ambient

4. Average operating temperature of fluid-filled system:

5. Ambient pressure range of operation:
   0 - 3000 psig

6. Overpressure or positive bias over ambient sea-water pressure, Δp or unpressurized:

   2 - 4 psi | 0 psi

   Rationale for pressurized versus unpressurized or the converse:

   Biased when moving shaft seals are present in compensated device

   Not biased when moving shaft seals are not present in compensated device

7. Design rationale for selection of this configuration of compensator as related to system requirements:

Performs function with minimum weight

8. Probable failure mode. Any redundancy to compensation, provision to be sea-water floodable as worst condition? Will system "fail-sick" or "fail-dead"?

- **Outward leakage of fluid with compensator is exhausted, then exchange of fluid and sea water.**	- **Exchange of fluid and sea water**
- **No redundancy**	- **Same**
- **Space available for a small amount of water to pool below active electrical parts.**	- **Same**

B. Materials selection for compensated system

Compensator housing or case material. Bare metal exterior or protective coating, paint?

Monel, Bare	**Rubber**

Heat transfer considerations, if any:

Temperature rises modest; no special provisions waterborne.

Elastomer type, physical properties, any MIL-SPEC? Supplier:

Buna N, no Mil Spec

If spring is used, what type, material, and is it exposed to sea water?

BeCu, yes

Use of external screens, baffles, or filters to minimize damage from sea-water particle contamination on elastomer or piston and internal housing. Give details, mesh size, or filtration:

NiCu, 10 x 10 Mesh

Marine fouling, corrosion protection:

Marine growth not present since vehicles are only Waterborne when actually in use.

(a) Paints and coatings, metal passivation

Devran paint on compensated enclosures.

(b) Galvanic protection

Proper choice of materials; no cathodic protection system present; plating and anodizing at unavoidable carbon steel-aluminum interfaces in motors.

(c) Use of protective greases, lubricants, bedding compounds

none

Estimated safety factor, materials life:

III. Compensating fluid considerations

A. Fluid selection

1. Type or trade name:

   Dow Corning DC-200 Silicone fluid, one centistoke

2. Properties of fluid, as related to operation of system:

   - **Electrically non-conducting**
   - **Void filler**
   - **Low viscosity**
   - **Small variation in viscosity with pressure and temperature**

3. Emulsifying or nonemulsifying; rationale for this decision:

   Non-emulsifying: preferable that minor intrusion of water remain separated in bottom of system.

4. Materials compatibility problems for compensated system? If so, cite problems and remedy.

   Elastomers must be selected to be compatible with fluid and sea water. Insulation must be selected to be compatible with fluid. Fluid exhibits strong solvent characteristic. Suitable materials are available.

B. Design Parameters for sizing of compensator

Please show sample calculations of design scheme on extra sheet.

Compensating fluid type Silicone fluid, Dow Corning DC-200 Viscosity One centistoke

Overall temp range 27-95 Waterborne ° F. 0-130 Tenderborne

Overall pressure range 0-4500 psig.

Was this compensator designed for a specific environmental range of operation or for the extremes of arctic and tropical conditions? Why?

Specification requirement

1. Compressibility factor, at depth ______ (bulk modulus effect).
   Est. Volume change - 3.9 %
2. Equipment storage, transportation
   Pressure Atmospheric
   Temp. range, ° F 0-130
   Est. volume change, heating ______ %
   Cooling ______ % } TTL 9.7 %
3. Operational Characteristics
   Fluid coefficient of thermal expansion 0.00075 /° F

   Surface Conditions
   Max. operating temp., ° F ______
   Max. ambient sea-water temp., ° F ______
   Est. volume change ______
   Depth Conditions
   Max. operating temp., ° F ______
   Lowest ambient sea-water temp., ° F ______
   Est. volume change ______

   } For design, max vol occurs at 130 F and 0 psig; min vol occurs at 27 F and 4500 psig. Difference is 7.8 % for temp and 3.9 % for pressure. These changes are additive, thus totaling 11.7 %.
4. Seal Leakage (est.) Allow for ~ 30 cc/hr
   Volume change for length of mission ~ 6 %
   Avg. shaft rpm or reciprocating rate 900-1800 RPM
   Type of seal, where used Carbon ring face when running

   For 100 hrs by spec; mission is ~ 8 hrs
5. Sea-water intrusion (est.) None cc/hr
   Where may it enter system?
   Volume change for length of mission ______ %
6. Other ______

Total usable compensator volume expressed as % of fluid volume in compensated system 20 % for motors, 14 % for distribution centers

Has this volume proved to be adequate for your system? Comment ______

Yes. Extreme environmental conditions used for design are not actually attained in service. See also para. V.C. below.

Total fluid volume in compensated system 1.5 ft^3 for motors, 9 ft^3 for distribution centers

IV. Special design requirements

A. Use of protective devices in compensated system. Enclose system diagram, if possible.

1. Sea-water leak detectors. Description, location.

Electrode(s) approximately 1/8 inch above lowest point of enclosure. Alarm at approximately 10^3 ohms from electrode to enclosure (or other electrode). Drain or sampling plug located close to electrode location.

2. Vent or relief valves

(a) Fluid expansion relief
Valve configuration, location, relief pressure:

5-8 psid, location near top of compensating piping

2-6 psid, location near top of compensting piping

(b) Electrical equipment gassing relief
Valve configuration, location, relief pressure:

No normal gassing for these systems.

(c) Valve discharge port protection from intrusion of sea water?
Sketch

None; valve operates in sea water.

3. Fluid volume indicators or device to monitor compensator volume:

Portable pressure gage; visual or tactile methods; all with vehicle tenderborne.

4. Temperature monitoring. Give type of sensor and location.

none

5. Differential pressure transducer (for Δp between compensating fluid and ambient sea-water pressure).

none

6. Other

B. Fluid circulation devices in rotating machinery to aid in cooling the compensating fluid. Any use of fluid filtering and/or cooling devices for electric motors, speed reducers, hydraulic pumps or motors, etc.

None necessary waterborne.

C. Sea water to compensating fluid seals in system

1. Type, MIL-SPEC No. (if known), material, where used, estimated leakage rate.

   (a) Static seals

   O-rings

   (b) Rotating or reciprocating seals

   Crane carbon ring face

   (c) Seal problems, remedy -

   Properly designed and executed, these types of seals are effective and trouble-free.

2. Use of protective devices for outboard shaft seals, including actuators:

   none

D. Fluid ports or penetrations, tube fittings, quick-disconnects

1. Give type, material, use in compensating system.

   MS type or quick disconnect type for almost all connections. All CRES.

2. For quick-disconnects, what type of protective caps or plugs are used to prevent sea-water intrusion?

   All quick disconnects are mated before vehicle is waterborne.

E. Other unique, desirable design features

V. Compensating system operation, maintenance

A. Physical system arrangement. Show diagram, identifying components.

Compensators are connected to their respective enclosures by pipe, flex hoses, and quick disconnects.

B. System fill, vent procedure

Quick disconnect at a high point is broken, and enclosures and compensator are separately filled and vented.

C. Predive positioning of compensator or final volume adjustment.
While compensators are designed for fluid expansion as well as contraction, as a practical matter they are often topped-off just prior to launching so as to increase available compensating margin. This may require bleeding after retreival in warm operating areas.

D. Predive and postdive compensating system checkout. Explain contamination removal, refilling, and venting procedure.

Pre-dive as in C above. Post-dive check for fluid loss is made.

E. Fluid change interval

F. Drain and flush procedures

G. Fluid reconditioning process, if any

Filtering to remove particulate material. Drying with silica gel.

H. For compensator overhaul, describe cleaning and reconditioning of metal and elastomeric parts - what cleaning solutions or solvents are used?

I. Compensator test methods, if any

VI. Applicable Documents

Manuals, MIL-SPECS, design standards, etc that were used as a reference in the design and fabrication of the compensating system.

Specification for building a research vehicle AUTEC I.

Fluid Pressure-Compensating Systems
Survey Information Sheet

Date: 1-29-71

Contributing Organization, address:
General Dynamics/Electric Boat Division
Eastern Point Road
Groton, Conn. 06340

Name, address, phone no. of contributor(s):

G. Lefoley, U.D.E., R & D Annex (203-446-2444)

I. Submersible name, or submersible system:

Seacliff and Turtle manipulator tool system

Designer (firm or organization):

General Dynamics/Electric Boat Division

Builder:

General Dynamics/Electric Boat Division

Design depth: Test depth:

6,500 ft

Operational date:

Description of fluid pressure-compensated system, general operational requirements of system: MANIPULATOR TOOL SYSTEM, pp. V-4 to V-5

The interchangeable tools (parallel jaw hand, scissors hand, cable cutter, and drill) have spring loaded piston type compensators to maintain positive 2-5 psi Δp above ambient.

II. Compensating device

A. Configuration (brief description):

Spring loaded piston

1. Special, or modification of existing hardware; supplier:

Special design by GD/EB for this application.

2. Drawing or sketch of device:

See attached sheet. (page V-58)

3. Ambient temperature range (design); what environmental considerations these temperature extremes represent:

The compensator is designed for an ambient thermal variation of 100° F. This represents a dockside minimum temp. of 0° F and a maximum ambient temp. of 100° F.

4. Average operating temperature of fluid-filled system:

It would be essentially equal to ambient sea water temperature.

5. Ambient pressure range of operation:

0-6500 ft

6. Overpressure or positive bias over ambient sea-water pressure, Δp or unpressurized:

2-5 psi above ambient

Rationale for pressurized versus unpressurized or the converse:

To prevent sea water leakage into unit

7. Design rationale for selection of this configuration of compensator as related to system requirements:

It was the simplest, and it satisfied our performance and configuration requirements.

8. Probable failure mode. Any redundancy to compensation, provision to be sea-water floodable as worst condition? Will system "fail-sick" or "fail-dead"?

The system would perform with a failure of the compensator, but the system would have to be cleaned and repaired upon surfacing of the vehicle.

B. Materials selection for compensated system

Compensator housing or case material. Bare metal exterior or protective coating, paint?

Cres casing with a 4-coat paint system.

Heat transfer considerations, if any:

Not applicable because there is very little heat generation in this equipment.

Elastomer type, physical properties, any MIL-SPEC? Supplier:

MIL-P-5516 AN6227 "O"-Ring Seals, Buna "N"

If spring is used, what type, material, and is it exposed to sea water?

Coil Spring
Exposed to Sea Water
Material: Cres.

Use of external screens, baffles, or filters to minimize damage from sea-water particle contamination on elastomer or piston and internal housing. Give details, mesh size, or filtration:

Not used

Marine fouling, corrosion protection:

(a) Paints and coatings, metal passivation

4-coat epoxy paint system

(b) Galvanic protection

Zinc anodes

(c) Use of protective greases, lubricants, bedding compounds

Lubriplate grease used on faying surfaces and on nuts, bolts and screws.

Estimated safety factor, materials life:

Not applicable

III. Compensating fluid considerations

A. Fluid selection

1. Type or trade name:

   Lubriplate No. 2 and MIL-H-5606B

2. Properties of fluid, as related to operation of system:

   Lubriplate proved to be good from a corrosion and lubrication standpoint.

3. Emulsifying or nonemulsifying; rationale for this decision:

   Non-emulsifying -- this was suitable for our application.

4. Materials compatibility problems for compensated system? If so, cite problems and remedy.

   none

B. Design Parameters for sizing of compensator

Please show sample calculations of design scheme on extra sheet.

Compensating fluid type Lubriplate No. 2 MIL-H-5606B Viscosity Spec. Standard

Overall temp range 0° F to 100° F ° F.

Overall pressure range 0 to 3000 psig.

Was this compensator designed for a specific environmental range of operation or for the extremes of arctic and tropical conditions? Why?

100° F temp. differential (question #3, pg. 2)

1. Compressibility factor, at depth 1.6 in.^3 (bulk modulus effect).
   Est. Volume change 1.6 %
2. Equipment storage, transportation
   Pressure ______
   Temp. range, ° F 0-100°
   Est. volume change, heating +1.35 %
   Cooling -3.1 %
3. Operational Characteristics
   Fluid coefficient of thermal expansion Spec. Std. /° F

   Surface Conditions
   Max. operating temp.,° F 100
   Max. ambient sea-water temp., ° F 85
   Est. volume change 1.35 %

   Depth Conditions
   Max. operating temp., ° F 28
   Lowest ambient sea-water temp., ° F 28
   Est. volume change -1.8
4. Seal Leakage (est.) 0 cc/hr
   Volume change for length of mission 0 %
   Avg. shaft rpm or reciprocating rate --
   Type of seal, where used --
5. Sea-water intrusion (est.) 0 cc/hr
   Where may it enter system?
   Volume change for length of mission 0 %
6. Other ______

Total usable compensator volume expressed as % of fluid volume in compensated system 6.1 %

Has this volume proved to be adequate for your system? Comment yes

Total fluid volume in compensated system 100 in.^3

IV. Special design requirements

A. Use of protective devices in compensated system. Enclose system diagram, if possible.

1. Sea-water leak detectors. Description, location.

None

2. Vent or relief valves **None**

(a) Fluid expansion relief
Valve configuration, location, relief pressure:

(b) Electrical equipment gassing relief
Valve configuration, location, relief pressure:

(c) Valve discharge port protection from intrusion of sea water?
Sketch

3. Fluid volume indicators or device to monitor compensator volume:

Visual observation of compensator piston displacement

4. Temperature monitoring. Give type of sensor and location.

none

5. Differential pressure transducer (for Δp between compensating fluid and ambient sea-water pressure).

none

6. Other

B. Fluid circulation devices in rotating machinery to aid in cooling the compensating fluid. Any use of fluid filtering and/or cooling devices for electric motors, speed reducers, hydraulic pumps or motors, etc.

none

C. Sea water to compensating fluid seals in system

1. Type, MIL-SPEC No. (if known), material, where used, estimated leakage rate.

   (a) Static seals

   MIL-P-5516 AN6227B Buna "N"

   (b) Rotating or reciprocating seals

   MIL-P-5516 AN6227 Buna "N"
   and REV- O-Slide Seals

   (c) Seal problems, remedy -

2. Use of protective devices for outboard shaft seals, including actuators:

   none

D. Fluid ports or penetrations, tube fittings, quick-disconnects

1. Give type, material, use in compensating system.

   none

2. For quick-disconnects, what type of protective caps or plugs are used to prevent sea-water intrusion?

   none

E. Other unique, desirable design features

none

V. Compensating system operation, maintenance

A. Physical system arrangement. Show diagram, identifying components.

See attached sheet

B. System fill, vent procedure

See attached sheet

C. Predive positioning of compensator or final volume adjustment.

See attached sheet

D. Predive and postdive compensating system checkout. Explain contamination removal, refilling, and venting procedure.

See attached sheet

E. Fluid change interval
Every 6 months or as required, dependent upon operational experience.

F. Drain and flush procedures

Drain and flush with freon cleaning agent.

G. Fluid reconditioning process, if any

none

H. For compensator overhaul, describe cleaning and reconditioning of metal and elastomeric parts - what cleaning solutions or solvents are used?

Parts are cleaned in an ultrasonic cleaner using freon as a cleaning agent.

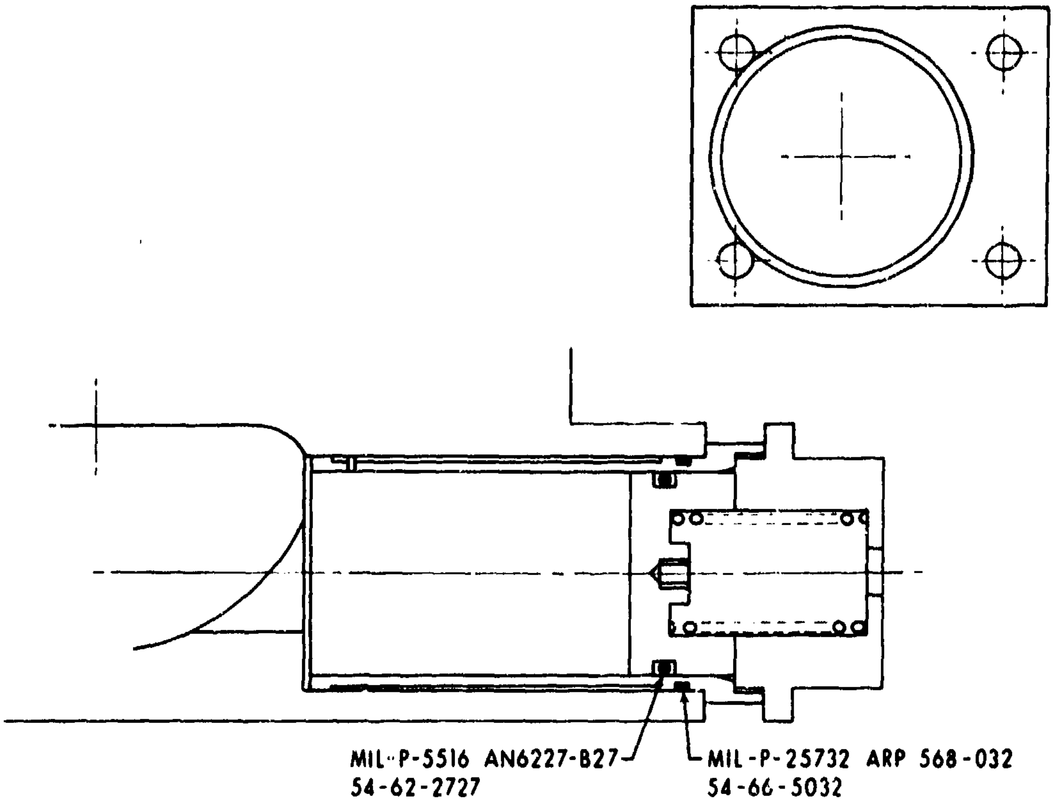
I. Compensator test methods, if any

none

VI. Applicable Documents

none

Manuals, MIL-SPECS, design standards, etc that were used as a reference in the design and fabrication of the compensating system.



Compensator Fill Instructions

It is necessary to preset the tool compensators for depth and temperature before each mission. Because the compensating chambers are small, the following filling procedures must be closely followed to prevent tool damage during operation.

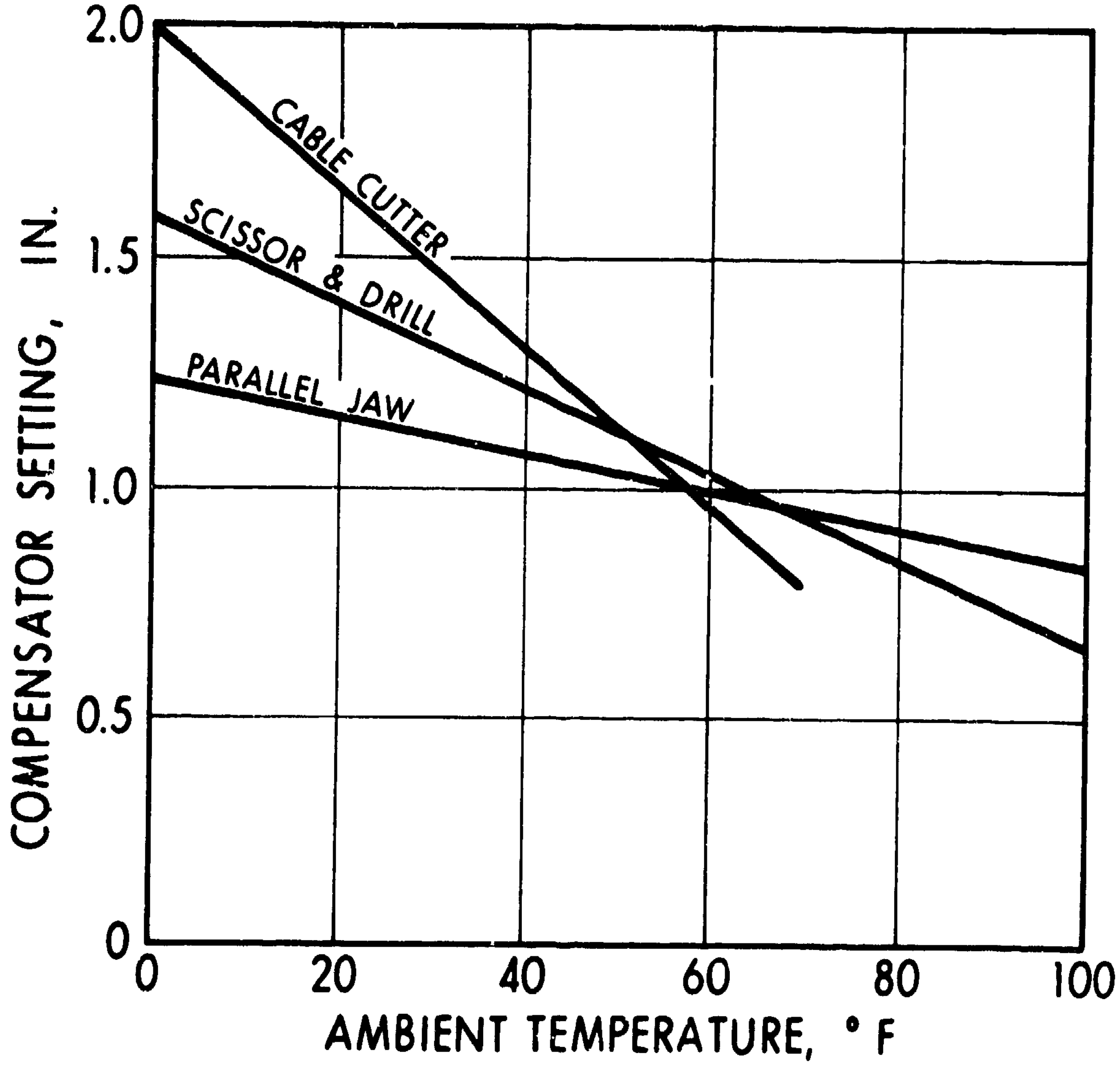
At the ambient temperature, check the (following) graph for the tool to obtain the compensator setting. This measurement is to be applied from the recess in the compensator piston to the outer edge of the piston retainer.

Note the following:

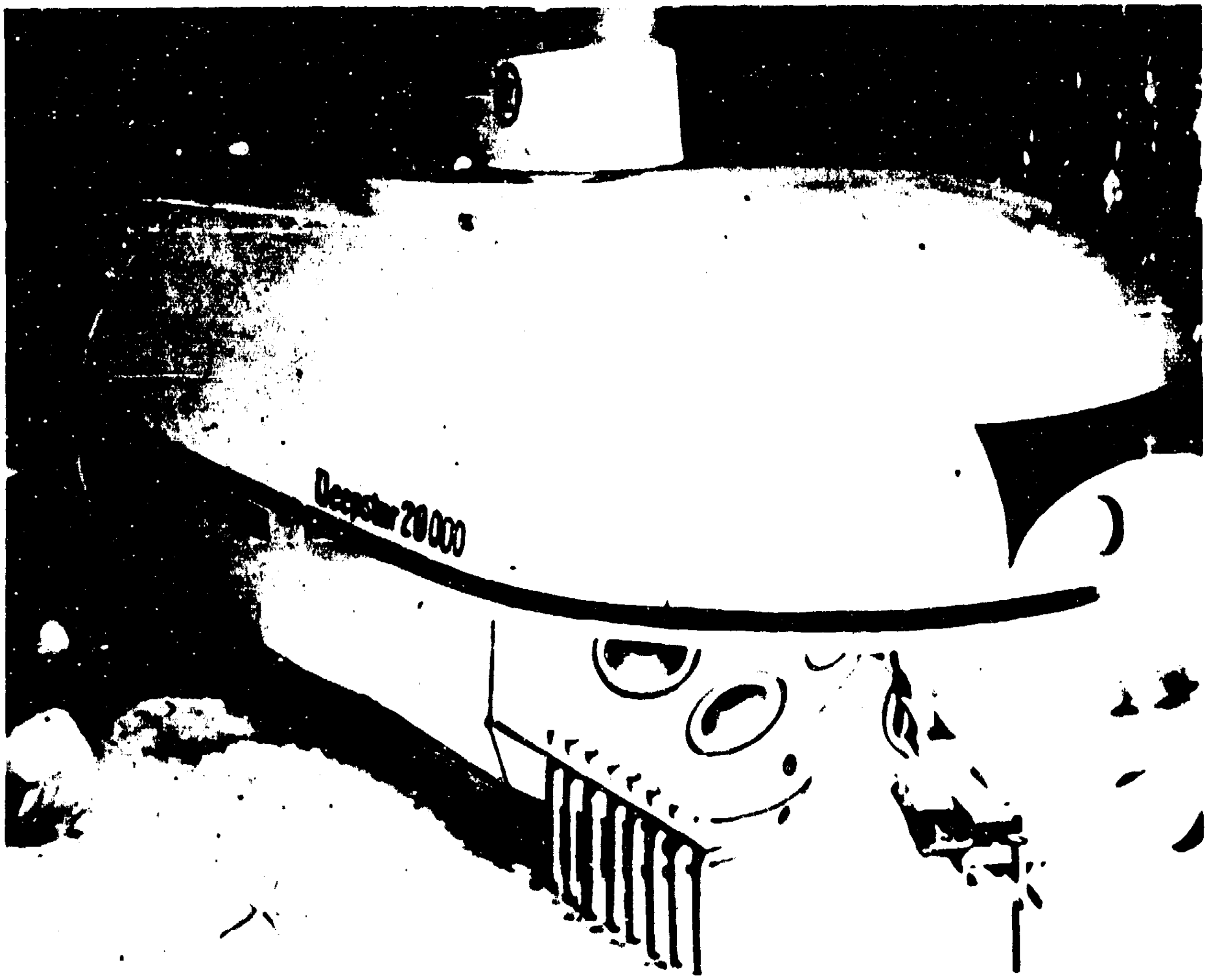
- If two compensators are used the graph value should be the average height of the pistons.
- Only deairiated oil should be used to fill the tools.
- Make certain all air is out of the tools.

• Do not allow cable cutter temperature to exceed 70° F.

• Fill cable cutter with MIL-H-5606 hydraulic fluid. Parallel jaw, scissor jaw, and drill are to be filled with Lubriplate 2.



5.3.3 DEEPSTAR 20,000 (construction deferred)



Fluid Pressure-Compensating Systems
Survey Information Sheet

Date: 3-31-71

Contributing Organization, address:

Westinghouse Ocean Research & Engineering Center
P.O. Box 1488, Annapolis, Md. 21404

Name, address, phone no. of contributor(s):

Charles B. Barclay, Jr. MS9850, 301-765-5510

I. Submersible name, or submersible system:

DEEPSTAR 20,000 Propulsion System Compensation

Designer (firm or organization):

Westinghouse

Builder:

Westinghouse

Design depth: **20,000 ft**

Test depth: **30,000 ft**

Operational date:
Not scheduled for completion.

Description of fluid pressure-compensated system, general operational requirements of system: VARIVEC PROPULSION SYSTEM, pp. V-61 to V-71

The DEEPSTAR 20,000 Propulsion Unit is the Westinghouse developed VARIVEC Propeller. The VARIVEC is a single drive unit designed to produce thrust vectors in any direction to propel and position the vehicle. The operating parts of the VARIVEC mechanism and the electric motor driving it are immersed in oil compensated to a pressure 2 to 10 psi above ambient sea water. This positive pressure is required to assure adequate sealing of the dynamic seals at the propeller hub and variable pitch blades.

II. Compensating device

A. Configuration (brief description):

Two identical piston type units provide the necessary compensation. Ambient pressure acts on each piston to compensate the oil within and a large coil spring in each unit provides the force needed for the positive differential.

1. Special, or modification of existing hardware; supplier:
Specially designed.

2. Drawing or sketch of device:

See attached sheet

3. Ambient temperature range (design); what environmental considerations these temperature extremes represent:

Ambient temperature range to be experienced is 28° F to 120° F for design purposes, assume:
1) Fill at 70° and lower to 28° F.
2) Fill at 70° and increase to 120° F.

4. Average operating temperature of fluid-filled system:

70° F (Estimate)

5. Ambient pressure range of operation:

1 ATM to 3100 psi

6. Overpressure or positive bias over ambient sea-water pressure, Δp or unpressurized:

Δ P = 2 to 10 psi depending upon spring extension.

Rationale for pressurized versus unpressurized or the converse:

A 2 to 10 psi positive pressure is required to assure adequate sealing of the dynamic seals for the hub and the variable pitch blades.

7. Design rationale for selection of this configuration of compensator as related to system requirements:

Minimum size and weight.

8. Probable failure mode. Any redundancy to compensation, provision to be sea-water floodable as worst condition? Will system "fail-sick" or "fail-dead"?

   1) **Leak in compensator unit: The compensators connect at the low point of the compensated system. Leakage in the compensators will result in loss of the positive Δ P. Since oil will remain in the mechanism, the system will continue to function.**
   2) **Leak near high point of system: A leak in a seal or hydraulic line at a high point in the system will result in sea water entry. The system will "Fail-Sick".**

B. Materials selection for compensated system

Compensator housing or case material. Bare metal exterior or protective coating, paint?

Alum Alloy 5086-H32
Bare metal exterior.

Heat transfer considerations, if any:

The compensated system is cooled by the ambient sea water surrounding it.

Elastomer type, physical properties, any MIL-SPEC? Supplier:

Rolling diaphragms - Buna - N Rubber with nylon reinforcement.
Bellofram Corp.
O-Rings - Material per MIL-P-25732 or MIL-P-5516.
Dimensions per MS28775.

If spring is used, what type, material, and is it exposed to sea water?

Coil Spring
Material: Tempered Alloy Steel
Immersed in oil.

Use of external screens, baffles, or filters to minimize damage from sea-water particle contamination on elastomer or piston and internal housing. Give details, mesh size, or filtration:

None Used.

Marine fouling, corrosion protection:

(a) Paints and coatings, metal passivation

Aluminum surfaces that require a smooth finish (for O-Ring Seals, etc.) are anodized per MIL-A-8625, Type III CL 1. All other surfaces are bare aluminum.

(b) Galvanic protection

None Used.

(c) Use of protective greases, lubricants, bedding compounds

Estimated safety factor, materials life:

Estimated life of rolling diaphragms: Two years.

III. Compensating fluid considerations

A. Fluid selection

1. Type or trade name:
   Brayco Micronic 762 Hydraulic Fluid.

2. Properties of fluid, as related to operation of system:

   Viscosity
   a) At 1 ATM Press, 100° F: 3.4 centistokes
   b) At 9050 psi, 32° F: 36 centistokes

3. Emulsifying or nonemulsifying; rationale for this decision:

   Emulsifying to reduce corrosion problems if sea water should enter the system.

4. Materials compatibility problems for compensated system? If so, cite problems and remedy.

   none

B. Design Parameters for sizing of compensator

Please show sample calculations of design scheme on extra sheet.

Compensating fluid type Bray 762 Hydraulic Fluid Viscosity See Sheet 5
Overall temp range 28° to 120° ° F.
Overall pressure range 0-9100 psig.

Was this compensator designed for a specific environmental range of operation or for the extremes of arctic and tropical conditions? Why?

1. Compressibility factor, at depth 2.9×10^5 (bulk modulus effect).
   Est. Volume change 3.14 %
2. Equipment storage, transportation
   Pressure 1 ATM
   Temp. range, ° F 28° to 120° F
   Est. volume change, heating 70° to 120° F +2.20%
   Cooling 70° to 28° F -1.85%
3. Operational Characteristics
   Fluid coefficient of thermal expansion 4.4×10^4 /° F

   Surface Conditions
   Max. operating temp., ° F 85° for Mech, 185° for motor
   Max. ambient sea-water temp., ° F 85°
   Est. volume change +3.1

   Depth Conditions
   Max. operating temp., ° F 70°
   Lowest ambient sea-water temp., ° F 28°
   Est. volume change -1.85
4. Seal Leakage (est.) Zero (Nominal) cc/hr
   Volume change for length of mission % blades reciprocate
   Avg. shaft rpm or reciprocating rate Hub 89 rpm. at 5°/sec
   Type of seal, where used Caterpillar ring seals for hub and blades
5. Sea-water intrusion (est.) Zero cc/hr
   Where may it enter system?
   Volume change for length of mission -- %
6. Other

Total usable compensator volume expressed as % of fluid volume in compensated system 795/4620 = 17.2%

Has this volume proved to be adequate for your system? Comment Yes
system has been tested at 9000 psi ambient.

Total fluid volume in compensated system 4620 IN^3

IV. Special design requirements

A. Use of protective devices in compensated system. Enclose system diagram, if possible.

1. Sea-water leak detectors. Description, location.

none

2. Vent or relief valves

(a) Fluid expansion relief
Valve configuration, location, relief pressure:

none

(b) Electrical equipment gassing relief
Valve configuration, location, relief pressure:

N.A.

(c) Valve discharge port protection from intrusion of sea water?
Sketch

N.A.

3. Fluid volume indicators or device to monitor compensator volume:

none

4. Temperature monitoring. Give type of sensor and location.

none

5. Differential pressure transducer (for Δp between compensating fluid and ambient sea-water pressure).

none

6. Other --

B. Fluid circulation devices in rotating machinery to aid in cooling the compensating fluid. Any use of fluid filtering and/or cooling devices for electric motors, speed reducers, hydraulic pumps or motors, etc.

Rotation of the mechanism parts circulates the compensating oil within the housing. The electric motor rotor circulates the oil within the motor housing. The entire propulsion unit is surrounded by ambient sea water.

C. Sea water to compensating fluid seals in system

1. Type, MIL-SPEC No. (if known), material, where used, estimated leakage rate.

   (a) Static seals

   MIL-P-25732 or MIL-P-5516

   (b) Rotating or reciprocating seals

   Caterpillar Ring Seals

   (c) Seal problems, remedy -

   Caterpillar Seals have proved to be satisfactory with 2 to 10 psi positive Δp.

2. Use of protective devices for outboard shaft seals, including actuators:

   none

D. Fluid ports or penetrations, tube fittings, quick-disconnects

1. Give type, material, use in compensating system.
   Tube fittings: 37° Flare, Parker, 316 Cres.
   Quick Disconnects: Snaptite 29 series, 316 Cres.

2. For quick-disconnects, what type of protective caps or plugs are used to prevent sea-water intrusion?

   316 Cres protective caps

E. Other unique, desirable design features

   --

V. Compensating system operation, maintenance

A. Physical system arrangement. Show diagram, identifying components.

Not available

B. System fill, vent procedure

Fill and bleed until all air is removed. Vacuum filling can not be used because of the rolling diaphragms.

C. Predive positioning of compensator or final volume adjustment.

System is filled to a specified pressure. This pressure is a function of the filling oil temperature.

D. Predive and postdive compensating system checkout. Explain contamination removal, refilling, and venting procedure.

Check compensator positions.
Take fluid samples.

E. Fluid change interval

When fluid samples indicate sea water content in excess of 2% , oil must be changed.

F. Drain and flush procedures

Drain andfill ports located at the low point of system allow for gravity drain. Bleed plugs are located at the high points. Pressure fill.

G. Fluid reconditioning process, if any

None at present.

H. For compensator overhaul, describe cleaning and reconditioning of metal and elastomeric parts - what cleaning solutions or solvents are used?

Solvent - Freon TF

I. Compensator test methods, if any

A pressure test to assure zero leakage is conducted at periodic maintenance intervals.

VI. Applicable Documents

Manuals, MIL-SPECS, design standards, etc that were used as a reference in the design and fabrication of the compensating system.

Ⓦ Spec 956A683 - Special Process Instruction, DEEPSTAR 20,000 Varivec Propulsion mechanism.

Ⓦ Spec 949A483 - DS 20 Hydraulic System Component General Spec.

MPR ASSOCIATES, INC.
1140 Connecticut Avenue, N.W.
Washington, D.C. 20036

Compensator Design

1. Compensation Volume Required

A. Pressure Contraction

(1) Bray Oil Bulk Modulus

$M_B = 2.9 \times 10^5$ psi   9100 psi

(2) $$\frac{\Delta v_P}{V} = \frac{\Delta P}{M_B} = \frac{9.1 \times 10^3}{2.9 \times 10^5} = 3.14 \times 10^{-2}$$

$$\frac{\Delta v_P}{V} = 3.14\%$$

B. Temperature Expansion/Contraction

(1) Bray Oil Coefficient of Expansion

$$\alpha = 4.4 \times 10^{-4} \ \frac{\text{in.}^3}{\text{in.}^3 \ {}^\circ\text{F}}$$

(2) Condition I

Assume total oil cavity volume varies from 120° F to 28° F

$\Delta T = 92°$ F

$$\frac{\Delta v_T}{V} = 4.4 \times 10^{-4} \times 92 = 4.05 \times 10^{-2}$$

$$\frac{\Delta v_T}{V} = 4.05\% \quad \text{or} \quad \left.\begin{matrix} +2.2\% \\ -1.85\% \end{matrix}\right\} \text{assuming fill at } 70^\circ \text{ F}$$

③ <u>Condition II</u>

Assume: Varivec @ 85° F
Motor @ 185° F

Assume: Varivec ≈ 13 gal
Motor ≈ 2 gal

Will analyze condition for motor running on surface in 85° F water and dive to +28° F (motor secured)

$$\Delta v_T = 4.4 \times 10^{-4} \left[(85 - 28)13 + (185 - 28)2\right]$$

$$= 4.4 \times 10^{-4} \left[13 \times 57 + 2 \times 157\right]$$

$$= 4.4 \times 10^{-4} [741 + 314]$$

$$\Delta v_T = 4.4 \times 10^{-4} [1055]$$

$$\frac{\Delta v_T}{V} = 4.4 \times 10^{-4} \frac{[1055]}{15} = 3.1 \times 10^{-2}$$

$$= 3.1\%$$

Therefore limiting condition from temperature standpoint is condition I.

C. Entrapped Air

Will assume that entrapped air accounts for 10% of the volume.

$$\frac{\Delta v_A}{V} = 10\%$$

D. Total Active Compensator Volume

1. $$\frac{\Delta v_{tot}}{V} = \frac{\Delta v_P}{V} + \frac{\Delta v_T}{V} + \frac{\Delta v_A}{V}$$

$$= 3.14 + 4.05 + 10 = 17.19\%$$

2. Assume mechanism volume in the order of 20 gal.

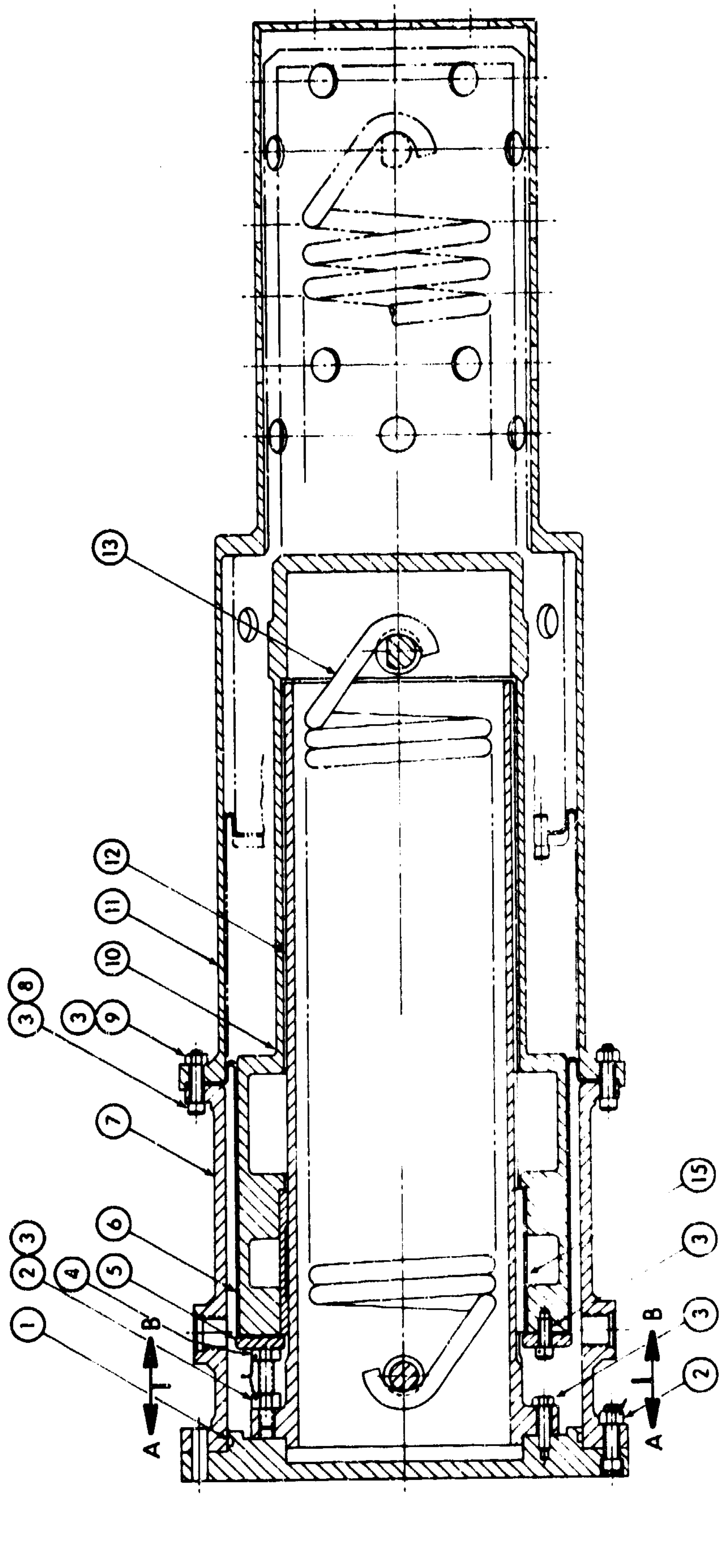
$$V_{mech} = 20 \text{ gal} \times 231 \text{ in.}^3/\text{gal} = 4620 \text{ in.}^3$$

$$\underline{V_{compensator} = 4620 \times 0.172 = 795 \text{ in.}^3}$$

3. Since the compensator volume is large should account for pressure and temperature changes in volume.

$$V_{comp.\ design} = 1.1 \times 795 \text{ in.}^3 = 875 \text{ in.}^3$$

$$= 3.79 \text{ gal.}$$



13
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Fluid Pressure-Compensating Systems
Survey Information Sheet

Date: 3-15-71

Contributing Organization, address:

Westinghouse Ocean Research and Engineering Center
P.O. Box 1488, Annapolis, Maryland 21404

Name, address, phone no. of contributor(s):

Charles B. Barclay, Jr. MS9850, 301-765-5510

I. Submersible name, or submersible system:

DEEPSTAR 20,000 Hydraulic and Electrical Component Compensation System

Designer (firm or organization):

Westinghouse

Builder:

Westinghouse

Design depth: 20,000 ft

Test depth: 30,000 ft

Operational date:
Not scheduled for completion.

Description of fluid pressure-compensated system, general operational requirements of system: HYDRAULIC & ELECTRICAL COMPONENTS, pp. V-76 to V-86

Hydraulic and electrical components that can not be exposed to sea water are protected by immersion in oil. This oil is contained by suitably sized enclosures. The compensating system described herein maintains the pressure of the oil within the enclosure at ambient pressure. Components protected by the system described include hydraulic solenoid valves, hydraulic pumps and motors, electric motors and electrical relays.

II. Compensating device

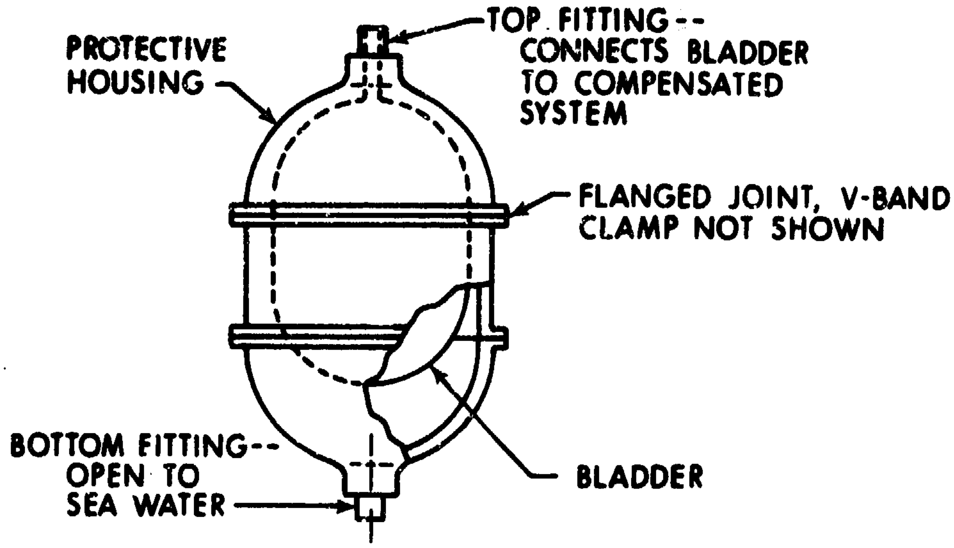
A. Configuration (brief description):

The compensating device consists of a suitably sized rubber bladder and a protective housing to prevent damage of the bladder.

1. Special, or modification of existing hardware; supplier:

Bladder - Standard Component. Greer Olaer, Inc.
Housing-Specially designed.

2. Drawing or sketch of device:



3. Ambient temperature range (design); what environmental considerations these temperature extremes represent:

   Ambient pressure range to be experienced is 28° F to 120° F.
   For design purposes, assume:
   1) Fill at 70° F and lower to 28° F.
   2) Fill at 70° F and increase to 120° F.

4. Average operating temperature of fluid-filled system:
   Enclosures containing valves, relays, etc.--ambient
   Enclosures containing motors, pump----100° F est.

5. Ambient pressure range of operation:

   1 ATM to 9050 psig

6. Overpressure or positive bias over ambient sea-water pressure, Δp or unpressurized:
   Unpressurized.
   If compensating bladder is located below the compensated unit a small +Δ P can be attained.

   Rationale for pressurized versus unpressurized or the converse:

   A positive Δ P system is always desirable but the additional weight required to accomplish it can not be justified in this case.

7. Design rationale for selection of this configuration of compensator as related to system requirements:

Minimum weight and size.

8. Probable failure mode. Any redundancy to compensation, provision to be sea-water floodable as worst condition? Will system "fail-sick" or "fail-dead"?

Probable failure mode: Leakage.
If the bladder is located below the compensated unit, bladder leakage will cause system to fail sick.
If leakage occurs in compensated unit, electrical shorting will occur.

B. Materials selection for compensated system

Compensator housing or case material. Bare metal exterior or protective coating, paint?

Housing - Fiberflass (GRP) and Plexiglass
Bladder - Buna - N Rubber
Fittings - 316 Cres and Nylon

Heat transfer considerations, if any:

Elastomer type, physical properties, any MIL-SPEC? Supplier:

Bladder - Buna N Rubber
O-Rings - Material per MIL-P-25732 or MIL-P-5516
Dimensions per MS28775

If spring is used, what type, material, and is it exposed to sea water?

N.A.

Use of external screens, baffles, or filters to minimize damage from sea-water particle contamination on elastomer or piston and internal housing. Give details, mesh size, or filtration:

--

Marine fouling, corrosion protection:

None - Units to be serviced and cleaned after each dive sequence.

(a) Paints and coatings, metal passivation

All Cres parts to be passivated per QQ-P-35 Type II

(b) Galvanic protection

None needed

(c) Use of protective greases, lubricants, bedding compounds

None needed

Estimated safety factor, materials life:

Estimated life of bladder: Two years

III. Compensating fluid considerations

A. Fluid selection

1. Type or trade name:

   Brayco Micronic 762 Hydraulic Fluid

2. Properties of fluid, as related to operation of system:

   Viscosity -
   a) At 1 ATM Press, 100° F: 3.4 centistokes
   b) At 9050 psi, 32° F: 36 centistokes
   SPG .855
   Flash Point, C.O.C. 210° F

3. Emulsifying or nonemulsifying; rationale for this decision:

   Emulsifying to reduce corrosion if sea water should enter the system.

4. Materials compatibility problems for compensated system? If so, cite problems and remedy.

   none

B. Design Parameters for sizing of compensator

Please show sample calculations of design scheme on extra sheet.

Compensating fluid type Bray 762 Hydraulic Fluid Viscosity See Sheet 5
Overall temp range 28° to 120° ° F.
Overall pressure range 0-9050 psig.

Was this compensator designed for a specific environmental range of operation or for the extremes of arctic and tropical conditions? Why?

Specific environmental range dictated by vehicle operating requirements.

1. Compressibility factor, at depth 2.9×10^5 (bulk modulus effect).
   Est. Volume change 3.12 % $\frac{9,050 \times 10^3}{2.9 \times 10^5} = 3.12$
2. Equipment storage, transportation
   Pressure 1 ATM
   Temp. range, ° F 28° to 120° F
   Est. volume change, heating 70° F to 120° F + 2.20% $4.4 \times 10^{-4} \times 50 = 2.20$
   Cooling 70° F to 28° F -1.85 % $4.4 \times 10^{-4} \times 42 = 1.85$
3. Operational Characteristics
   Fluid coefficient of thermal expansion 4.4×10^{-4} /° F

   Surface Conditions
   Max. operating temp., ° F 120°
   Max. ambient sea-water temp., ° F 85°
   Est. volume change +1.54 %

   Depth Conditions
   Max. operating temp., ° F 80° (EST)
   Lowest ambient sea-water temp., ° F 28°
   Est. volume change -2.28 %
4. Seal Leakage (est.) 0.1 cc/hr
   Volume change for length of mission -- %
   Avg. shaft rpm or reciprocating rate
   Type of seal, where used O-Ring face seals ~~O-Ring gland seals~~
5. Sea-water intrusion (est.) 0.1 cc/hr
   Where may it enter system? Shaft Seals
   Volume change for length of mission -- %
6. Other Entrapped air in oil, assume 10 %

Total usable compensator volume expressed as % of fluid volume in compensated system 700/2850 = 24.5 %

Has this volume proved to be adequate for your system? Comment
This volume will be more than adequate for the system.
It allows for margin in event of leakage from the system.
Total fluid volume in compensated system $2850\ IN^3$ (MAX) **per compensator unit. Three such units are utilized on the vehicle to compensate three isolated systems.**

IV. Special design requirements

A. Use of protective devices in compensated system. Enclose system diagram, if possible.

1. Sea-water leak detectors. Description, location.

A sea water detector is located at the low point of each compensated container. The detector consists of two brass plates of approximately 1.0 sq. in. area. Plate separation is 0.35 in. and voltage across the plates is 28 V.D.C. With electronic amplification, the device can detect a 2 % (Approx.) sea water emulsion.

2. Vent or relief valves

(a) Fluid expansion relief

Valve configuration, location, relief pressure:

A 10 psi pop-off relief valve is located near each bladder compensator unit.

(b) Electrical equipment gassing relief

Valve configuration, location, relief pressure:

--

(c) Valve discharge port protection from intrusion of sea water?

Sketch

The relief valve is a pop-off type (Circle Seal Co.) which minimizes possibility of sea water entry.

3. Fluid volume indicators or device to monitor compensator volume:

none

4. Temperature monitoring. Give type of sensor and location.

none

5. Differential pressure transducer (for Δp between compensating fluid and ambient sea-water pressure).

none

6. Other

--

B. Fluid circulation devices in rotating machinery to aid in cooling the compensating fluid. Any use of fluid filtering and/or cooling devices for electric motors, speed reducers, hydraulic pumps or motors, etc.

The oil within the hydraulic pump/electric motor enclosure is circulated by the motor rotor. The enclosure walls are in direct contact with sea water which cools the oil within.

C. Sea water to compensating fluid seals in system

1. Type, MIL-SPEC No. (if known), material, where used, estimated leakage rate.

   (a) Static seals

   MIL-P-25732 or MIL-P-5516

   (b) Rotating or reciprocating seals

   MIL-P-25732 or MIL-P-5516

   (c) Seal problems, remedy -

   ---

2. Use of protective devices for outboard shaft seals, including actuators:

   Wipers

D. Fluid ports or penetrations, tube fittings, quick-disconnects

1. Give type, material, use in compensating system.

   Tube fittings: 37° Flare, Parker, 316 Cres.
   Quick Disconnects: Snaptite 29 series, 316 Cres.

2. For quick-disconnects, what type of protective caps or plugs are used to prevent sea-water intrusion?

   316 Cres Protective Caps

E. Other unique, desirable design features

V. Compensating system operation, maintenance

A. Physical system arrangement. Show diagram, identifying components.

Not available

B. System fill, vent procedure

Fill and bleed until all air is removed. Vacuum filling can not be used because of rubber bladder.

C. Predive positioning of compensator or final volume adjustment.

System is filled with a specified amount of fluid. Amount is dependent upon ambient temperature.

D. Predive and postdive compensating system checkout. Explain contamination removal, refilling, and venting procedure.

Prior to each dive sequence, system is checked to insure proper filling. After each dive sequence, bladders are visually checked to insure proper operation.

E. Fluid change interval

When fluid samples indicate sea water content in excess of 2 %, oil must be changed.

F. Drain and flush procedures

Quick disconnect fittings allow for hook-up of hydraulic servicing unit to fill system. System is gravity drained.

G. Fluid reconditioning process, if any

None at present.

H. For compensator overhaul, describe cleaning and reconditioning of metal and elastomeric parts - what cleaning solutions or solvents are used?

Solvent - Freon TF

I. Compensator test methods, if any

A 5 psi leak check is conducted at periodic maintenance intervals. Acceptable leakage rate: Zero

VI. Applicable Documents

Manuals, MIL-SPECS, design standards, etc that were used as a reference in the design and fabrication of the compensating system.

Ⓦ Spec 949A483 - DS 20 Hydraulic System Component General Spec.

Fluid Pressure-Compensating Systems
Survey Information Sheet

Date: 3-18-71

Contributing Organization, address:

Westinghouse Ocean Research and Engineering Center
P.O. Box 1488, Annapolis, Maryland 21404

Name, address, phone no. of contributor(s):

Charles B. Barclay, Jr., MS 9850, 301-765-5510

I. Submersible name, or submersible system:

DEEPSTAR 20,000 Hydraulic System (Working Fluid)

Designer (firm or organization):

Westinghouse

Builder:

Westinghouse

Design depth: 20,000 ft

Test depth: 30,000 ft

Operational date:
Not scheduled for completion.

Description of fluid pressure-compensated system, general operational requirements of system: HYDRAULIC SYSTEM (working fluid), pp. V-87 to V-97

The DS 20,000 Hydraulic System consists basically of three pumps, two reservoirs, four valve packages, and various actuators and hydraulic motors that drive the working functions (manipulator, recovery winch, weight droppers, etc). The reservoirs function to maintain the system return pressure at 50 psi above ambient (System operating). The pumps are variable displacement units that maintain supply line pressure at 2800 to 3200 psi above reservoir pressure. Valve configuration is such that the lines downstream of the valves are pressure compensated when the valves are closed.

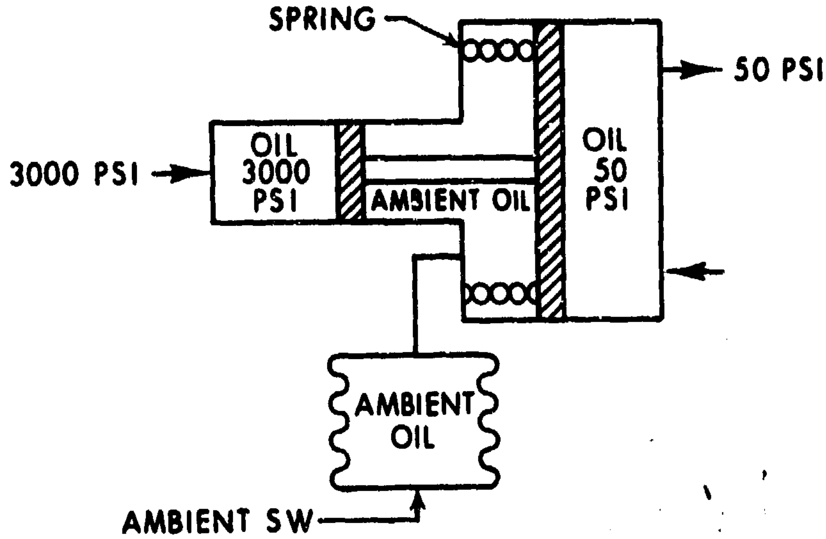
II. Compensating device

A. Configuration (brief description):

The reservoir is a piston/cylinder device. Ambient pressure acts on the piston to compensate the fluid within. In addition, the reservoir maintains return fluid at 50 psi above ambient (System operating) or 1 to 5 psi above ambient (System secured).

1. Special, or modification of existing hardware; supplier:
Modification of existing hardware.
Supplier: Pneudraulics, Inc., Montclair, Calif.

2. Drawing or sketch of device:



3. Ambient temperature range (design); what environmental considerations these temperature extremes represent:

Ambient temperature range to be experienced is 28° to 120° F. for design purposes, assume:
1) Fill at 70° and Lower to 28° F.
2) Fill at 70° and Increase to 120° F.

4. Average operating temperature of fluid-filled system:

100° F (Estimate)

5. Ambient pressure range of operation:

1 ATM to 9050 psig

6. Overpressure or positive bias over ambient sea-water pressure, Δp or unpressurized:

ΔP = 1 to 5 psi (System unpressurized)
ΔP = 50 psi (System pressurized)

Rationale for pressurized versus unpressurized or the converse:

A positive pressure bias of 50 psi is required to:
1) Prevent pump cavitation (when operating at or near surface)
2) Improve sealing capability of O-Ring seals.
3) Minimize opportunity for sea water intrusion.

7. Design rationale for selection of this configuration of compensator as related to system requirements:

Minimum weight and size.

8. Probable failure mode. Any redundancy to compensation, provision to be sea-water floodable as worst condition? Will system "fail-sick" or "fail-dead"?

Probable failure cause: Leakage. If system return develops a leak, the reservoir piston will bottom and sea water will enter the system. Since the hydraulic system will operate for a limited time with a sea water/oil mixture as hydraulic fluid, the system will "Fail Sick".

If system supply line leaks, system will "Fail Sick" or "Fail Dead" depending on magnitude of leak.

B. Materials selection for compensated system

Compensator housing or case material. Bare metal exterior or protective coating, paint?

Housing Material: 356-T6 Aluminum Anodized per MIL-A-8625 and painted.
Paint: Polyurethane (Laminar X-500)

Heat transfer considerations, if any:

Reservoir surrounded by sea water.

Elastomer type, physical properties, any MIL-SPEC?
Supplier: **Bellows (to compensate reservoir)-BUNA-N/Nylon**
O-Rings - Material per MIL-P-25732 or MIL-P-5516.
Dimensions per MS28775.

If spring is used, what type, material, and is it exposed to sea water?

Coil Spring, 17-7PH
Spring exposed to oil, not sea water.

Use of external screens, baffles, or filters to minimize damage from sea-water particle contamination on elastomer or piston and internal housing. Give details, mesh size, or filtration:

A rubber bellows type boot filled with oil protects the ambient pressure side of the piston from sea water. Refer to schematic on Page 2.

Marine fouling, corrosion protection:

(a) Paints and coatings, metal passivation

Aluminum exposed to sea water is anodized per MIL-A-8625 and painted with laminar X-500 polyurethane paint.

(b) Galvanic protection

none

(c) Use of protective greases, lubricants, bedding compounds

none

Estimated safety factor, materials life:

Estimated life of unit: Two years

III. Compensat g fluid considerations

A. Fluid selection

1. Type or trade name:

Brayco Micronic 762 Hydraulic Fluid

2. Properties of fluid, as related to operation of system:

Visoosity
a) At 1 ATM press, 100° F: 3.4 centistokes
b) At 12000 psi, 32° F: 54 centistokes
SPG .855
Flash Point, C.O.C. 210° F.

3. Emulsifying or nonemulsifying; rationale for this decision:

Emulsifying to reduce corrosion problems if sea water should enter system.

4. Materials compatibility problems for compensated system? If so, cite problems and remedy.

none

B. Design Parameters for sizing of compensator

Please show sample calculations of design scheme on extra sheet.

Compensating fluid type Bray 762 Hydraulic Fluid Viscosity See Sheet 5

Overall temp range 28° F to 120 ° F.

Overall pressure range 0-12100 psig.

Was this compensator designed for a specific environmental range of operation or for the extremes of arctic and tropical conditions? Why?

Specific environmental range dictated by vehicle operating requirements.

1. Compressibility factor, at depth 2.9×10^5 (bulk modulus effect).

   Est. Volume change 4.17 % $\frac{12.10\times10^3}{2.9\times10^5} = 4.17$ %

2. Equipment storage, transportation

   Pressure 1 ATM

   Temp. range, ° F 28° to 120°

   Est. volume change, heating 70° F to 120° F +2.20 %

   Cooling 70° F to 28° F -1.85 %

3. Operational Characteristics

   Fluid coefficient of thermal expansion 4.4×10^{-4} /° F

   Surface Conditions

   Max. operating temp.,° F 120° (Shut down if temp. exceeds 120° F)

   Max. ambient sea-water temp., ° F 85

   Est. volume change +1.54 %

   Depth Conditions

   Max. operating temp., ° F 80° EST

   Lowest ambient sea-water temp., ° F 28°

   Est. volume change -2.28 %

4. Seal Leakage (est.) 3.0 cc/hr

   Volume change for length of mission 0.2 %

   Avg. shaft rpm or reciprocating rate --

   Type of seal, where used --

5. Sea-water intrusion (est.) Zero cc/hr

   Where may it enter system?

   Volume change for length of mission -- %

6. Other Assume entrapped air accounts for 7% of the volume. (System is vacuum filled.)

   Total usable compensator volume expressed as % of fluid volume in compensated system 17 %

   Has this volume proved to be adequate for your system? Comment

   Depth testing of system has not been performed.

   Total fluid volume in compensated system 2000 IN^3

IV. Special design requirements

A. Use of protective devices in compensated system. Enclose system diagram, if possible.

1. Sea-water leak detectors. Description, location.

A sea water detector is located in the return line for each of the two hydraulic reservoirs. The detector consists of two brass plates of approximately 1.0 sq. in. in area. Plate separation is 0.35 in. and voltage across the plates is 28 V.D.C. With electronic amplification the device can detect a 2 %(Approx.) sea water emulsion.

2. Vent or relief valves
   (a) Fluid expansion relief
   Valve configuration, location, relief pressure:

   A 70 psi pop-off relief valve is located in the return line immediately downstream of one of the two hydraulic reservoirs. Operating pressure for the reservoirs is 50 psi.

   (b) Electrical equipment gassing relief
   Valve configuration, location, relief pressure:

   N.A.

   (c) Valve discharge port protection from intrusion of sea water?
   Sketch

   The relief valve is a pop-off type which minimizes possibility of sea water entry.

3. Fluid volume indicators or device to monitor compensator volume:

A switch is used to indicate when the reservoirs are at low level (1/4 full). The switch is a magnetic reed switch encased in a hard cylinder. The actuating magnet rides on the reservoir piston.

4. Temperature monitoring. Give type of sensor and location.

None Used

5. Differential pressure transducer (for Δp between compensating fluid and ambient sea-water pressure).

None Used

6. Other ---

B. Fluid circulation devices in rotating machinery to aid in cooling the compensating fluid. Any use of fluid filtering and/or cooling devices for electric motors, speed reducers, hydraulic pumps or motors, etc.

None

C. Sea water to compensating fluid seals in system

1. Type, MIL-SPEC No. (if known), material, where used, estimated leakage rate.

   (a) Static seals

   MIL-P-25732 or MIL-P-5516

   (b) Rotating or reciprocating seals

   MIL-P-25732 or MIL-P-5516

   (c) Seal problems, remedy -

   --

2. Use of protective devices for outboard shaft seals, including actuators:

   Wipers

D. Fluid ports or penetrations, tube fittings, quick-disconnects

1. Give type, material, use in compensating system.

   Tube fittings - 37° Flare, Parker, 316 Cres
   Quick Disconnects - Snaptite 29 series, 316 Cres.

2. For quick-disconnects, what type of protective caps or plugs are used to prevent sea-water intrusion?

   316 Cres Protective Caps.

E. Other unique, desirable design features

--

V. Compensating system operation, maintenance

A. Physical system arrangement. Show diagram, identifying components.

Not available

B. System fill, vent procedure

Vacuum Fill

C. Predive positioning of compensator or final volume adjustment.

Reservoir must be completely filled prior to dive.

D. Predive and postdive compensating system checkout. Explain contamination removal, refilling, and venting procedure.

**Check level of sea water contamination. Drain, flush, and refill system if sea water contamination exceeds 2 %.
Change filters at end of each dive sequence.**

E. Fluid change interval
Drain, flush, and refill system during each overhaul or if sea water contamination exceeds 2 %.

F. Drain and flush procedures

Drain plugs at low points of system allow for gravity draining of most oil. In addition, supply and return line service connections with quick disconnects allow for flushing of system with external power supply.

G. Fluid reconditioning process, if any

None at present

H. For compensator overhaul, describe cleaning and reconditioning of metal and elastomeric parts - what cleaning solutions or solvents are used?

Solvent - FREON TF

I. Compensator test methods, if any

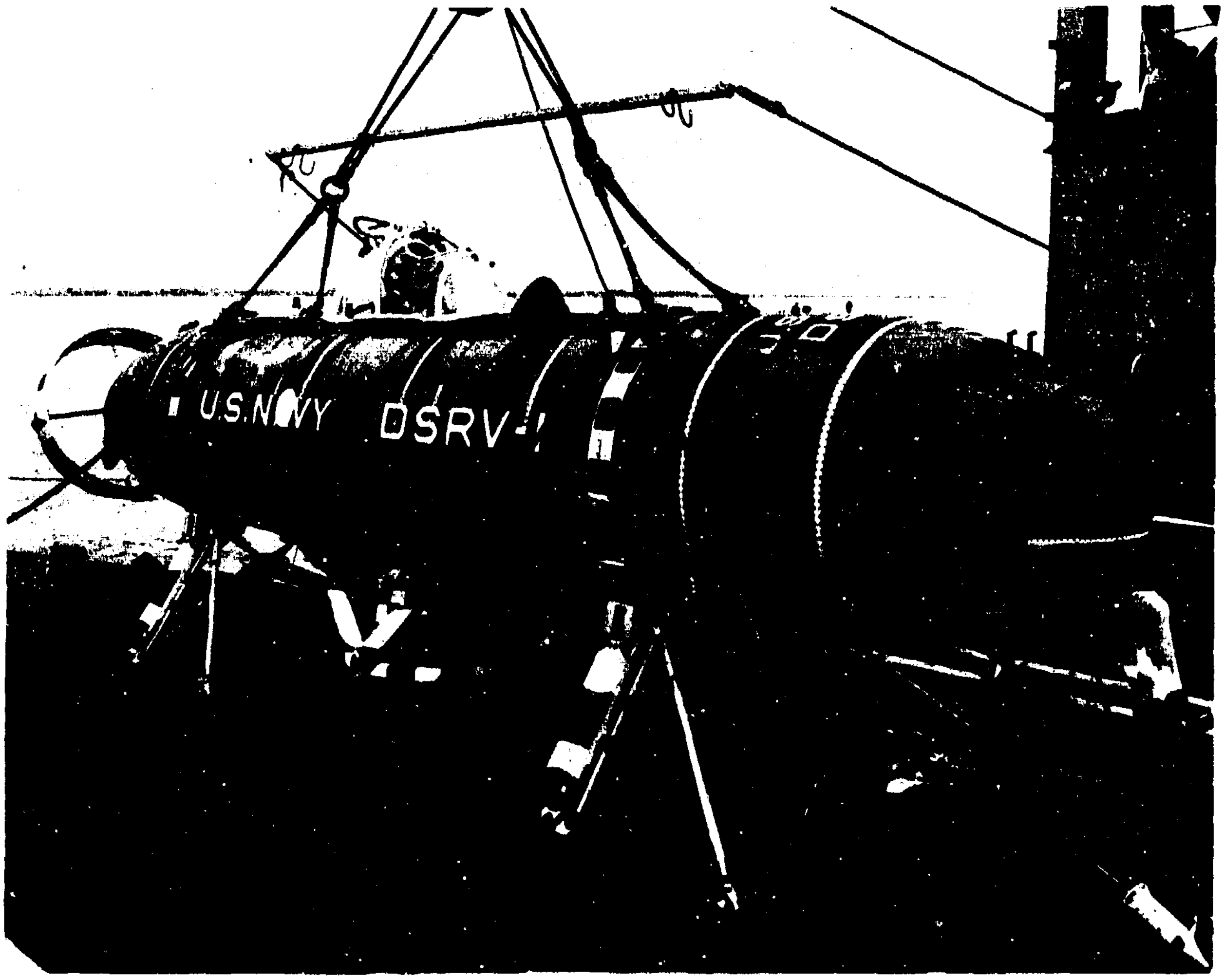
A proof pressure test to assure zero leakage is conducted at periodic maintenance intervals.

VI. Applicable Documents

Manuals, MIL-SPECS, design standards, etc that were used as a reference in the design and fabrication of the compensating system.

Ⓥ Spec 949A483 - DS 20 Hydraulic System Component General Spec.

5.3.4 DSRV-I



Enclosure (4)

Fluid Pressure-Compensating Systems
Survey Information Sheet

Date: 2-18-71

Contributing Organization, address:
Lockheed Missiles and Space Company
P.O. Box 504
Sunnyvale, California

Name, address, phone no. of contributor(s):
E.P. Pallange
0/57-20, B/150
(408) 742-7526

I. Submersible name, or submersible system:

Deep Submergence Rescue Vehicle (DSRV)

Designer (firm or organization):
Lockheed Missiles and Space Company
Sunnyvale, California

Builder:

Lockheed Missiles and Space Company

Design depth: 5000 ft

Test depth: 3600 ft

Operational date:
1971

Description of fluid pressure-compensated system, general operational requirements of system:

External systems which cannot be directly exposed to seawater because of electric continuity problems, material corrosion or fluid lubricity but are not significantly affected by the high compressive pressures, are compensated with oil.

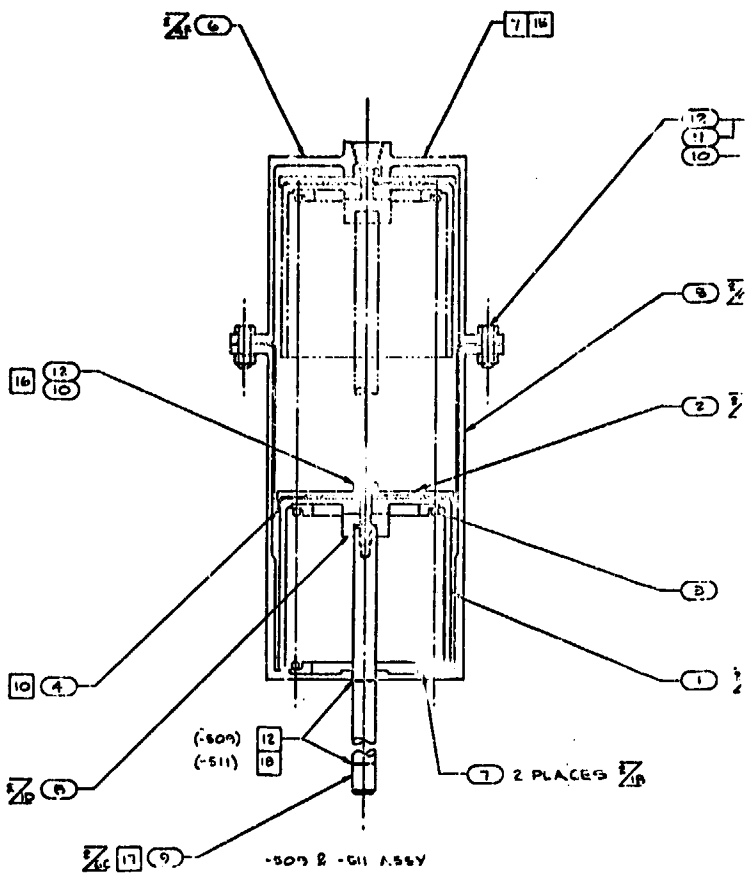
II. Compensating device

A. Configuration (brief description):

The device is cylindrical in shape with a spring loaded flexible elastomeric separator (diaphragm).

1. Special, or modification of existing hardware; supplier:

2. Drawing or sketch of device:



3. Ambient temperature range (design); what environmental considerations these temperature extremes represent:

-40° to + 140° F

Air transportable requirement and exposure to on board surface ship, arctic and tropic conditions.

4. Average operating temperature of fluid-filled system:

28° F to 100° F

5. Ambient pressure range of operation:

Sea level to 5000 foot depths.

6. Overpressure or positive bias over ambient sea-water pressure, Δp or unpressurized:

+3 psid (pounds per square inch differential)

Rationale for pressurized versus unpressurized or the converse:

Prevent seepage of seawater into compensated system.

7. Design rationale for selection of this configuration of compensator as related to system requirements:

Because of the many systems to be compensated with varying requirements, two sizes were selected and are used in varied groupings to provide required compensating capacity.

8. Probable failure mode. Any redundancy to compensation, provision to be sea-water floodable as worst condition? Will system "fail-sick" or "fail-dead"?

Loss of compensating fluid due to system leakage. After loss of positive pressure seawater must still replace compensating oil before a major equipment breakdown occurs. Therefore, regular system volume indicator checks will minimize probability of total compensator fluid loss due to leakage. In addition, many compensated items have seawater indicators to warn of the presence of seawater in compensated equipment.

B. Materials selection for compensated system

Compensator housing or case material. Bare metal exterior or protective coating, paint?

Aluminum alloy 6061-T6 type painted with a polyurethane (see enclosure (9).

Heat transfer considerations, if any:

Elastomer type, physical properties, any MIL-SPEC? Supplier:

Bellofram - see enclosed dwg. 301-2921055 (page II-8).

If spring is used, what type, material, and is it exposed to sea water?

Coil, K-Monel per QQ-N-286 Class A age hardened - exposed to sea-water

Use of external screens, baffles, or filters to minimize damage from sea-water particle contamination on elastomer or piston and internal housing. Give details, mesh size, or filtration:

none used

Marine fouling, corrosion protection:

(a) Paints and coatings, metal passivation

Painted per Spec. RV-S-0110. See enclosure (9)

(b) Galvanic protection -

(c) Use of protective greases, lubricants, bedding compounds

none

Estimated safety factor, materials life: -

III. Compensating fluid considerations

A. Fluid selection

1. Type or trade name:
   MIL-H-6083C Hydraulic Oil. See enclosure (2).
   MIL-S-21568 Silicone 5 centistoke. See enclosure (3).
2. Properties of fluid, as related to operation of system:

   Silicone more suitable for electrical switching.
   Hydraulic better lubricity than silicone.

3. Emulsifying or nonemulsifying; rationale for this decision:

   Basically non-emulsifying

4. Materials compatibility problems for compensated system? If so, cite problems and remedy.

   No specific problems

B. Design Parameters for sizing of compensator

Please show sample calculations of design scheme on extra sheet.

Compensating fluid type see Enclosure (2)(3) Viscosity ________

Overall temp range -40° to +160° ° F.

Overall pressure range vacuum to 4000 psig psig.

Was this compensator designed for a specific environmental range of operation or for the extremes of arctic and tropical conditions? Why?

1. Compressibility factor, at depth Enclosures (2) and (3) (bulk modulus effect).
   - Est. Volume change ________ %
2. Equipment storage, transportation
   - Pressure 9.3 to 15.1 psia
   - Temp. range, ° F -40 to 140°
   - Est. volume change, heating N/A %
   - Cooling N/A %
3. Operational Characteristics
   - Fluid coefficient of thermal expansion Enclosures (2) and (3) /° F
   - Surface Conditions
     - Max. operating temp., ° F 85
     - Max. ambient sea-water temp., ° F 85
     - Est. volume change N/A*
   - Depth Conditions
     - Max. operating temp., ° F ________
     - Lowest ambient sea-water temp., ° F 28
     - Est. volume change N/A
4. Seal Leakage (est.) N/A cc/hr
   - Volume change for length of mission ________ %
   - Avg. shaft rpm or reciprocating rate ________
   - Type of seal, where used ________
5. Sea-water intrusion (est.) N/A cc/hr
   - Where may it enter system?
   - Volume change for length of mission ________ %
6. Other ________

Total usable compensator volume expressed as % of fluid volume in compensated system Enclosure (8) %

Has this volume proved to be adequate for your system? Comment Operation too limited to date for comment.

Total fluid volume in compensated system Enclosure (8)

*N/A = Not Available

IV. Special design requirements

A. Use of protective devices in compensated system. Enclose system diagram, if possible.

1. Sea-water leak detectors. Description, location.

Most electrical boxes have seawater detectors

2. Vent or relief valves

(a) Fluid expansion relief
Valve configuration, location, relief pressure:

Yes, see dwg 2652742

(b) Electrical equipment gassing relief
Valve configuration, location, relief pressure:

See dwg 2652742 for relief valve

(c) Valve discharge port protection from intrusion of sea water?
Sketch

See dwg 2659344 - Standard off the shelf valve

3. Fluid volume indicators or device to monitor compensator volume:

See dwg 2652633. Protruding rod used.

4. Temperature monitoring. Give type of sensor and location.

None used

5. Differential pressure transducer (for Δp between compensating fluid and ambient sea-water pressure).

None used

6. Other --

B. Fluid circulation devices in rotating machinery to aid in cooling the compensating fluid. Any use of fluid filtering and/or cooling devices for electric motors, speed reducers, hydraulic pumps or motors, etc.

No specific provisions are made except in the hydraulic system where filtration is provided as part of the basic system and fluid circulation results from normal flow patterns.

C. Sea water to compensating fluid seals in system

1. Type, MIL-SPEC No. (if known), material, where used, estimated leakage rate.

   (a) Static seals

   Standard MS type O-rings or similar O-ring type seals with material suitable for either the MIL-H-6083 or MIL-S-21568 fluids.

   (b) Rotating or reciprocating seals

   Data not immediately available.

   (c) Seal problems, remedy -

   None known to date

2. Use of protective devices for outboard shaft seals, including actuators:

D. Fluid ports or penetrations, tube fittings, quick-disconnects

1. Give type, material, use in compensating system.

   Synflex nylon tubing manufactured by Samuel Moore Co., Mantua, Ohio, used for all basic piping.

2. For quick-disconnects, what type of protective caps or plugs are used to prevent sea-water intrusion?

   Caps and plugs as appropriate, designed for use with the QD's are used on every QD

E. Other unique, desirable design features

V. Compensating system operation, maintenance

A. Physical system arrangement. Show diagram, identifying components.

See dwg 2652742

B. System fill, vent procedure

See enclosed preliminary procedure, enclosure (6)

C. Predive positioning of compensator or final volume adjustment.

Only system visual inspection

D. Predive and postdive compensating system checkout. Explain contamination removal, refilling, and venting procedure.

A predive and post dive visual inspection of all compensators is made for proper volume indication and indication of any fluid leakage.

E. Fluid change interval

See enclosure (6)

F. Drain and flush procedures

See enclosure (6)

G. Fluid reconditioning process, if any

Not recommended

H. For compensator overhaul, describe cleaning and reconditioning of metal and elastomeric parts - what cleaning solutions or solvents are used?

See enclosure (6).

I. Compensator test methods, if any

See enclosures (6) and (7), also note that assembly testing is specified (ref dwg 2652633).

VI. Applicable Documents

Manuals, MIL-SPECS, design standards, etc that were used as a reference in the design and fabrication of the compensating system.

See enclosures.

DSRV-I Survey Enclosures

(1) Chart 1, List of Compensated Equipment on DSRV

(2) & (3) Technical Reports (OMITTED)

(4) Compensating Systems Survey (see page V-100)

(5) Equipment Drawings (OMITTED)

(6) Preliminary Service Procedure

(7) Standard Operating Procedure - Check for Entrapped Air in Compensated Systems, SMP 11

(8) DSRV Compensated Systems Volumes

(9) Specification RV-S-0110 (OMITTED)

CHART 1

LIST OF COMPENSATED EQUIPMENT

ITEM	COMPENSATED EQUIPMENT	QTY	FLUID	COMPENSATOR	QTY/EQUIPMENT
1	Main Battery Assembly	2	Marcol 70	Large	3
2	Vertical Thruster	2	MIL-H-6083C	Small	1
3	Horizontal Thruster	2	MIL-H-6083C	Small	1
4	Cable Junction Box	2	MIL-S-21568	Small	1
5	Power Distribution Box	2	MIL-S-21568	Large	2
6	Shore Power Interlock Box	1	MIL-S-21568	Small	1 1/2 *
7	Midbody Cable Junction Box	1	MIL-S-21568	Small	1 1/2 *
8	Main Propulsion Motor	1	MIL-H-6083C	Small	2
9	Hydraulic Power Unit	2	MIL-H-6083C	Integral	-

Small Compensator - P/N 301-2652633

Large Compensator - P/N 301-2652632

*Items 6 and 7 each have one small compensator and share one.

Enclosure (6)

PMS DATA INPUT FORM

NON-CERTIFIABLE ITEMS

Contract NObs 63(A)

Contractor LMSC Mfg. Date 11-13-70

Equipment Name Compensator Systems Part No. 2652742

Safety Precautions

See attached sheet.

Tools, Parts, Materials & Test Equipment Required

See attached sheet.

Procedure | Periodicity

A. Clean and Inspect Compensators and Related Systems — After Return to Operational Base

1. Clean components as follows:

a) Remove access panels to gain access to compensator and related system. Refer to Table 1 for access covers and locations.

WARNING: Wear safety goggles while working with compressed air.

b) Blow all dirt and foreign matter from the compensator and lines, tubing, and fittings using dry air at a pressure of 50 psig.

c) Use a clean, lint-free cloth and soap (Specification P-S-600) to wipe clean all accessible components, tubing, and fittings.

WARNING: Use only MIL-C-81302, Type 1, solvent. Do not use any other hydrocarbon, chlorinated, or alcoholic-type solvents on or near any equipment used with the DSRV. Introduction of minute residues of these other materials into the DSRV life support system can result in severe illness or death.

Safety Precautions

WARNING

Use only MIL-C-81302, Type 1, solvent. Do not use any other hydrocarbon, chlorinated, or alcoholic-type solvents on or near any equipment used with the DSRV. Introduction of minute residues of these other materials into the DSRV life support system can result in severe illness or death.

WARNING

Wear safety goggles while working with compressed air.

WARNING

Use solvent only in a well-ventilated area; avoid breathing solvent vapor. Keep flame and other sources of ignition away from solvent and solvent vapors.

WARNING

High pressures are required to test operation of the valves. Failure to observe the following safety procedures could result in serious injury.

Install a relief valve downstream of pressure regulator and set it to 110% of the maximum specified test pressure.

Secure all flexible lines to prevent whipping.

Mount valve securely behind a suitable safety shield.

Apply test pressures carefully. After 50% of maximum test pressure is reached, slowly increase pressure in increments of 10%. Hold pressure for 2 min. between each increase to allow valve to stabilize.

Hold maximum test pressure for at least 5 min. before conducting visual examination. Use mirrors to observe unit whenever possible.

Unless otherwise specified, pneumatic test pressures that may expose personnel to injury must not exceed 25% of design burst pressure.

Tools, Parts, Materials & Test Equipment

Tools
Safety goggles
Stiff-bristled brush
Open-end wrenches (5/16, 7/16, 7/8, 5/8, and 1-5/16 in.)
Crowsfoot (7/8, 1, and 1-5/16 in. for 1/4-in. drive)
Torque wrench (0 to 150 in.-lb)
Socket (1/2 in.)

Replacement Parts
Diaphragm (P/N 2652638-1 for compensator P/N 2652633) (1 reqd per compensator)
Diaphragm (P/N 2652636-1 for compensator P/N 2652632) (1 reqd per compensator)
O-rings (P/N 8180-011 and 8180-114 for globe valve P/N 2650351-7)
(1 ea. reqd per valve)
Relief valve (P/N 2659344-1) (13 reqd)

Materials
Fresh water supply
50-psi compressed air
Soap (Specification P-S-600)
Clean, lint-free cloth
Solvent (Specification MIL-C-81302, Type 1)
Aluminum-oxide abrasive cloth (400 grit)
Hydraulic fluid (Specification MIL-H-6083)

Test Equipment
Compensating service units (P/N 2651041-503, -505, and -507)
Hydraulic test cart
Pressure source (4,000 psig)

Table 1

Location and Identification of Compensators and Fill Connectors

Equipment Being Compensated	Fill Connector Access: Door Location (Station/Angle)	Fill Connector Access: Connector Identification Number	Compensator(s) Access: Door Location (Station/Angle)	Compensator(s) Access: Compensator Identification Number	Compensating Service Unit Part No./Operating Fluid
Forward Vertical Thruster Motor	55/185	P33	32/257	P12	2651041-503/MIL-H-6083
Forward Horizontal Thruster Motor	55/185	P34	32/257	P13	2651041-503/MIL-H-6083
Forward Cable Junction Box	55/185	P36	32/315	P18	2651041-505/MIL-S-21568, 5 Centistoke
Forward Main Battery Box	126/180	P32	153/15 153/315 153/315	P21 P19 P20	2651041-507/Marcol 70, 13.5 Centistoke
Forward Power Distribution Box	126/180	P35	153/283 153/285	P22 P23	2651041-505/MIL-S-21568, 5 Centistoke
Shore Power Interlock Box	237/180	P37	219/350 237/285	P25[(a)] P47[(a)]	2651041-505/MIL-S-21568, 5 Centistoke
Midbody Cable Junction Box	237/180	P38	219/350 237/285	P24[(a)] P47[(a)]	2651041-505/MIL-S-21568, 5 Centistoke
After Power Distribution Box	405/180	P39	405/283 405/283	P29 P30	R651041-505/MIL-S-21568, 5 Centistoke
After Main Battery Box	405/180	P40	405/77 405/77 405/345	P27 P28 P26	2651041-507/Marcol 70, 13.5 Centistoke
After Horizontal Thruster Motor	522/270	P43	522/270	P15	2651041-503/MIL-H-6083

(a) Shore power interlock box and midbody junction box have a common compensator as well as an individual one for each item.

Table 1 (Cont'd)

Location and Identification of Compensators and Fill Connectors

Equipment Being Compensated	Fill Connector Access: Door Location (Station/Angle)	Fill Connector Access: Connector Identification Number	Compensator(s) Access: Door Location (Station/Angle)	Compensator(s) Access: Compensator Identification Number	Compensating Service Unit Part No./Operating Fluid
After Vertical Thruster Motor	522/270	P42	522/270	P14	2651041-503/MIL-H-6083
After Cable Junction Box	522/270	P41	522/270	P31	2651041-505/MIL-S-21568, 5 Centistoke
Main Propulsion Motor	522/270	P44	555/45 555/315	P16 P17	2651041-503/MIL-H-6083

PMS DATA INPUT FORM

Procedure	Periodicity

WARNING: Use solvent only in a well-ventilated area; avoid breathing solvent vapor. Keep flame and other sources of ignition away from solvent and solvent vapors.

d) Use a clean, lint-free cloth moistened with solvent (Specification MIL-C-81302) to remove grease and oil.

e) Use a stiff-bristled brush or aluminum-oxide abrasive cloth (400 grit) to loosen encrusted residue and to clean difficult to reach areas.

f) Rinse all components using fresh water and dry using a clean, lint-free cloth or dry, low-pressure air.

2. Inspect compensators as follows:

a) Check fluid level in each compensator per NAVSHIPS 0905-120-2010 "DSRV System Checkout and Servicing." Refill systems as necessary.

b) Inspect all nylon hoses and fittings for leakage and deterioration. Replace defective components.

B. <u>Remove, Disassemble, Clean, Reassemble, Test, and Install Compensators</u> — <u>Every 18 Mo.</u>

1. Remove compensators (P/N 2652632 and 2652633) from DSRV as follows:

a) Plug Compensating Service Unit listed in Table 1 into the fill connection of the system from which the compensator is to be removed.

b) Operate the pump so that it draws the compensating fluid from the system to the tank.

c) Disconnect Compensating Service Unit and discard oil removed from system.

d) Loosen bolts that secure the compensator to its mounting bracket.

e) Disconnect hydraulic lines at the back of the compensator. Plug or cap lines.

f) Remove mounting bolts and the compensator from the vehicle.

g) Store mounting bolts in a clean polyethylene bag. Identify and retain for reinstallation.

h) Remove the remaining bolts in flange around periphery of compensator. Separate cover plate and body of compensator.

Procedure | Periodicity

2. Disassemble compensators as follows:

   a) Remove bolt securing the center of diaphragm in the compensator.

   b) Remove cover plate over the diaphragm.

   c) Remove diaphragm and internal components. Discard diaphragm.

   NOTE: Disassembly of the large and small compensators is identical, except as follows:

Compensator Part No.	Diaphragm Part No.	Wrench Size
2652632	2652636-1	5/3 in.
2652633	2652638-1	7/16 in.

   d) Inspect all parts for excessive wear, dents, corrosion, cracks, stripped or crossed threads, and other damage.

3. Clean all reusable compensator components per Step A.1.

4. Reassemble compensators as follows:

   CAUTION: Replacement parts must meet all design and specification requirements of the part being replaced.

   a) Assemble compensator using new diaphragm. (See Step 2 for Part No. of diaphragms.)

   b) Assemble compensators per Drawings 2652632 and 2652633 for large and small compensators, respectively.

   c) Tighten bolts that secure center of diaphragm to compensator rod to 45 to 55 in.-lb torque.

   d) Tighten bolts around flange of compensator to 60 to 65 in.-lb torque.

5. Test compensators as follows:

   a) Fill each unit with fluid as specified in Table 1.

   b) Pressurize compensator to 4.0 ± 0.2 psi. After 1 hr at this pressure there must be no visible evidence of leakage.

   c) Bleed pressure off slowly and record pressure at beginning and end of rod travel and stroke.

   d) Repressurize and repeat cycle 10 times. Movement on each cycle must be smooth, with no jamming. Stroke must be 9.1 ± 0.2 in. and pressure at each end of stroke must be as follows:

Procedure	Periodicity

Rod extended - 1.5 ± 0.3 psi
Rod retracted - 0.5 ± 0.1 psi.

6. Install compensators in DSRV as follows:

   a) Support the compensator in the position from which it was removed in Step 1.

   b) Install mounting bolts and nuts removed in Step 1 through the compensator flange and the mounting bracket. Tighten nuts to 60 to 65 in.-lb torque.

   c) Connect the hydraulic lines to the back of the compensator and tighten the fittings to 35 to 40 in.-lb torque.

C. <u>Remove, Disassemble, Clean, Reassemble, Test, and Install Vent Connections</u> — <u>Every 18 Mo.</u>

NOTE: Each system has a vent connection, located in the vehicle as shown in Table 2.

1. Perform preventive maintenance on quick-disconnect couplings (P/N 2652KM8 and 2653KM8, Seaton & Wilson Mfg. Co., Inc.) as follows:

   a) Remove coupling from system.

   b) Disassemble, clean, inspect, reassemble, and test coupling per PMS general procedure "Quick Disconnect-Coupler and Nipple" dated August 1970.

   c) Install coupling in system and tighten to 35 to 40 in.-lb torque.

2. Perform preventive maintenance on manually operated globe valve (P/N 2650351-7) as follows:

   a) Cut lockwire and remove handle. (See Fig. 1.)

   b) Remove locknut and bonnet.

   c) Remove stem poppet from bonnet.

   d) Remove poppet from stem.

   e) Inspect all parts for excessive wear, damage, or corrosion.

   f) Remove O-rings from the bonnet and poppet. Destroy O-rings.

   WARNING: Use only MIL-C-81302, Type 1, solvent. Do not use any other hydrocarbon, chlorinated, or alcoholic-type solvents on or near any equipment used with the DSRV. Introduction of minute residues of these other materials into the DSRV life support system can result in severe illness or death.

Table 2

Location and Identification of Compensator Vent Connectors & Relief Valves

Equipment Being Compensated	Vent Connector Access		Relief Valve (P/N 2659344-1) Access	
	Door Location Station/Angle	Connector Identification Number	Door Location Station/Angle	Relief Valve Identification Number
Forward Vertical Thruster Motor	32/320	P49	32/320	P68
Forward Horizontal Thruster Motor	32/320	P48	32/320	P69
Forward Cable Junction Box	32/320	P83	32/40	P66
Forward Main Battery Box(a)	88/0	P67	88/0	P65
Forward Power Distribution Box	88/0	P84	88/0	P64
Shore Power Interlock Box	237/298	P85	237/298	P70
Midbody Cable Junction Box	237/298	P86	237/298	P63
After Power Distribution Box	467/0	P88	467/0	P73
After Main Battery Box(a)	467/0	P75	467/0	P72
After Horizontal Thruster Motor	522/270	P59	555/315	P62
After Vertical Thruster Motor	522/270	P58	555/315	P61
After Cable Junction Box	522/270	P87	555/315	P60
Main Propulsion Motor	522/270	P89	555/315	P71

(a) All vent connectors are nipples for quick disconnect couplings P/N 2659261-11, except for those vents for the batteries that are manually operated Glove valves P/N 2650351-7.

PMS DATA INPUT FORM

Procedure | Periodicity

WARNING: Wear safety goggles while working with compressed air.

WARNING: Use solvent only in a well-ventilated area; avoid breathing solvent vapor. Keep flame and other sources of ignition away from solvent and solvent vapors.

g) Flush and clean valve bonnet, body, and poppet with solvent (Specification MIL-C-81302, Type 1). Dry parts using 50 psi compressed air.

h) Inspect all parts for cracks, dents, corrosion, stripped or crossed threads, deformation, and other obvious damage. Install new parts if damaged beyond minor repair.

CAUTION: Replacement parts must meet all design and specification requirements of the part being replaced.

i) Install new O-rings on bonnet and poppet per PMS general procedure "O-Ring Removal, Inspection, and Replacement" dated June 1970.

j) Reassemble valves in reverse order of disassembly. (See Fig. 1.)

k) Test valves as follows:

WARNING: High pressures are required to test operation of the valves. Failure to observe the following safety procedures could result in serious injury.

Install a relief valve downstream of pressure regulator and set it to 110% of the maximum specified test pressure.

Secure all flexible lines to prevent whipping.

Mount valve securely behind a suitable safety shield.

Apply test pressures carefully. After 50% of maximum test pressure is reached, slowly increase pressure in increments of 10%. Hold pressure for 2 min. between each increase to allow valve to stabilize.

Hold maximum test pressure for at least 5 minutes before conducting visual examination. Use mirrors to observe unit whenever possible.

Unless otherwise specified, pneumatic test pressures that may expose personnel to injury must not exceed 25% of design burst pressure.

Bleed off pressure in test setup after completion of tests.

Procedure	Periodicity

1) Apply 150 ± 5 psig pressure to valve; flow rate at this pressure must be 5 gpm.

2) When pressures from zero to 225 psig are applied to either port, the external leakage must be zero when the valve is exposed to an ambient pressure of 1 atm.

3) With zero pressure at one port and the valve closed, internal leakage must be zero when valve is exposed to ambient pressure of 4000 psig.

l) Inform maintenance officer if valve fails to pass tests.

m) If valve operation is satisfactory, install valve in DSRV in the same position from which it was removed.

D. Replace Relief Valves — Every 18 Mo.

NOTE: Each compensator system has a relief valve (P/N 2659344-1), located in the vehicle as shown in Table 2.

1. Remove relief valve from system and discard it.

   CAUTION: Replacement parts must meet all design and specification requirements of the part being replaced.

2. Install a new relief valve, tightening connections to 35 to 45 in.-lb torque.

E. Fill and Bleed Compensator Systems — Every 18 Mo.

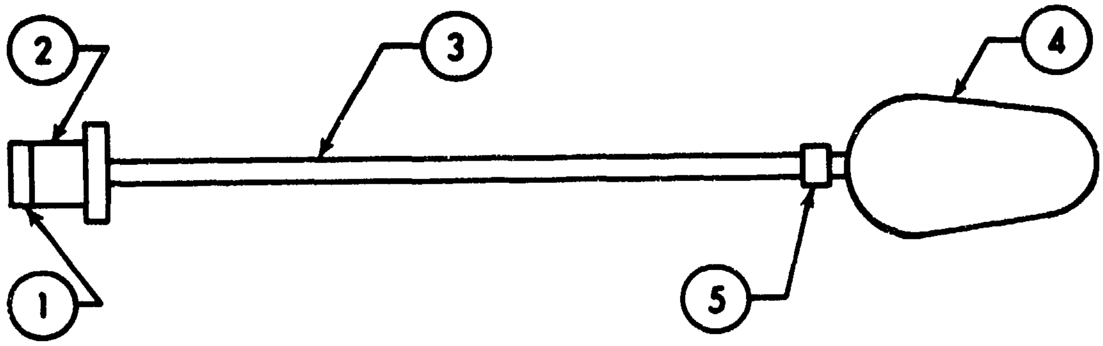
1. Connect Compensating Service Unit (see Table 1) to the system to be filled.

2. Connect bleed hose assembly (P/N 2651590-309) to the vent connection when filling the system for the main propulsion motor. Use bleed hose assembly (P/N 2651590-311) when filling the system for the thruster motors. Use bleed hose assembly shown in Fig. 2 when filling all other systems.

3. Fill compensator system until all air has been bled and indicator on compensator shows full. Refer to PMS general procedure "Drain, Fill, and Vent Fluid Systems" dated July 1970 for proper methods.

4. Disconnect fill and vent equipment.

5. Check system for leaks by inspecting indicator on compensator approximately 30 minutes after filling system. If indicator has retained its position, the system is not leaking. If indicator has moved the system leaks. Inform maintenance officer if leakage cannot be stopped by tightening a loose fitting or repairing obvious defects.



Item	Part No.	Description	Supplier
1	2659261-17	Dust Plug	LMSC
2	2659261-13	Quick-Disconnect Coupling	LMSC
3	2A06PW45A10H-0360	Flex Hose Assy	Preece, Inc.
4	2651590-1	Bladder Assy	LMSC
5	2651590-3	Adapter	LMSC

Fig. 2 Bleeder Hose Assembly for Filling Compensator Systems (Excluding Main Propulsion Motor and Thruster Motors)

Enclosure (7)
SMP No. 11
Revision Original
Page 1

STANDARD MAINTENANCE PROCEDURE

Check for Entrapped Air in Compensated Systems

A. Components and/or Systems Affected

	System	Fluid	Connector Fill	Connector Relief	Maximum Allowable Air
1.	Fwd Horiz. Thruster Motor	6083	P-34	P-69	4.3 in^3
2.	Fwd Vert Thruster Motor	6083	P-33	P-68	4.3 in^3
3.	Fwd J-Box	Sil Oil	P-36	P-66	18.0 in^3
4.	Fwd Main Battery	Marcol	P-32 or P-45	P-65	14.60 in^3
5.	Fwd Power Distribution Box	Sil Oil	P-35	P-64	208.0 in^3
6.	Shore Power I/L Box	Sil Oil	P-37	P-70	29.5 in^3
7.	Mid Body J-Box	Sil Oil	P-38	P-63	18.2 in^3
8.	Trim & List Systems	6083	L-6	L-18	302.0 in^3
9.	Aft Power Distribution Box	Sil Oil	P-39	P-73	208.0 in^3
10.	Aft Main Battery	Marcol	P-40 or P-46	P-72	146.0 in^3
11.	Aft J-Box	Sil Oil	P-41	P-60	18.2 in^3
12.	Aft Horiz Thruster Motor	6083	P-43	P-62	4.3 in^3
13.	Aft Vert Thruster Motor	6083	P-42	P-61	4.3 in^3
14.	Main Propulsion Motor	6083	P-44	P-71	11.2 in^3

B. Special Precautions

1. Verify correct compensating fluid is being used for system being checked and that it is clean.
2. Do not exceed 5.0 psig pressure on systems; damage to compensators may result.
3. Verify that system has been properly filled in accordance with applicable procedure.

C. Special Servicing Equipment

1. Source of clean compensating fluid per A above and Figure 1.
2. Pressure gauge 0 to 10 psig per Figure 1.
3. Two stop valves, (1) and (2) per Figure 1.
4. Hose, tubing, fittings and Q.D. for fill connections per Figure 1.

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Revision Original
Page 2

D. Procedure

1. For the system being checked;
   - gag the relief valve indicated in A above.
   - connect the equipment set-up indicated in Figure 1 to the Fill Quick Disconnect indicated in A above.

2. With valves (1) and (2) open,pressurize the system to 4 ±0 psig (18.7 psia) and close valve (1).

3. Inspect system to see that
   - compensator indicator rod(s) are all the way out (third groove on the rod visible).
   - gagged relief valve is not leaking.
   - no other leaks are visible.

4. Verify that there has been no significant change in the system pressure (no change greater than 0.1 psig).

5. Crack valve (2), catching the fluid in the catch vessel per Figure 1, bleed down the pressure to 3 ±0 psig (17.7 psia), and shut valve (1).

6. Determine volume of oil in catch vessel in cubic inches.

7. Determine actual volume of air in system by formula:

   Volume of air = 18.7 x volume of oil to drop pressure 1 psia

8. Compare calculated volume of air with the "allowable" volume indicated in A above:
   - Actual air volume ____________ in^3
   - Allowable air volume ____________ in^3

9. If actual air volume exceeds the "allowable" go back to appropriate fill procedure for further vacuum-fill or repeat fill and bleed for the system in question. Then repeat this SMP.

Prepared by: J H Terry 10/8/70
J. H. Terry Date

Approvals: LMSC [signature] 10/8/70
R. H. Bleakley, Manager Date
Test Operations

NAVY ____________
Date

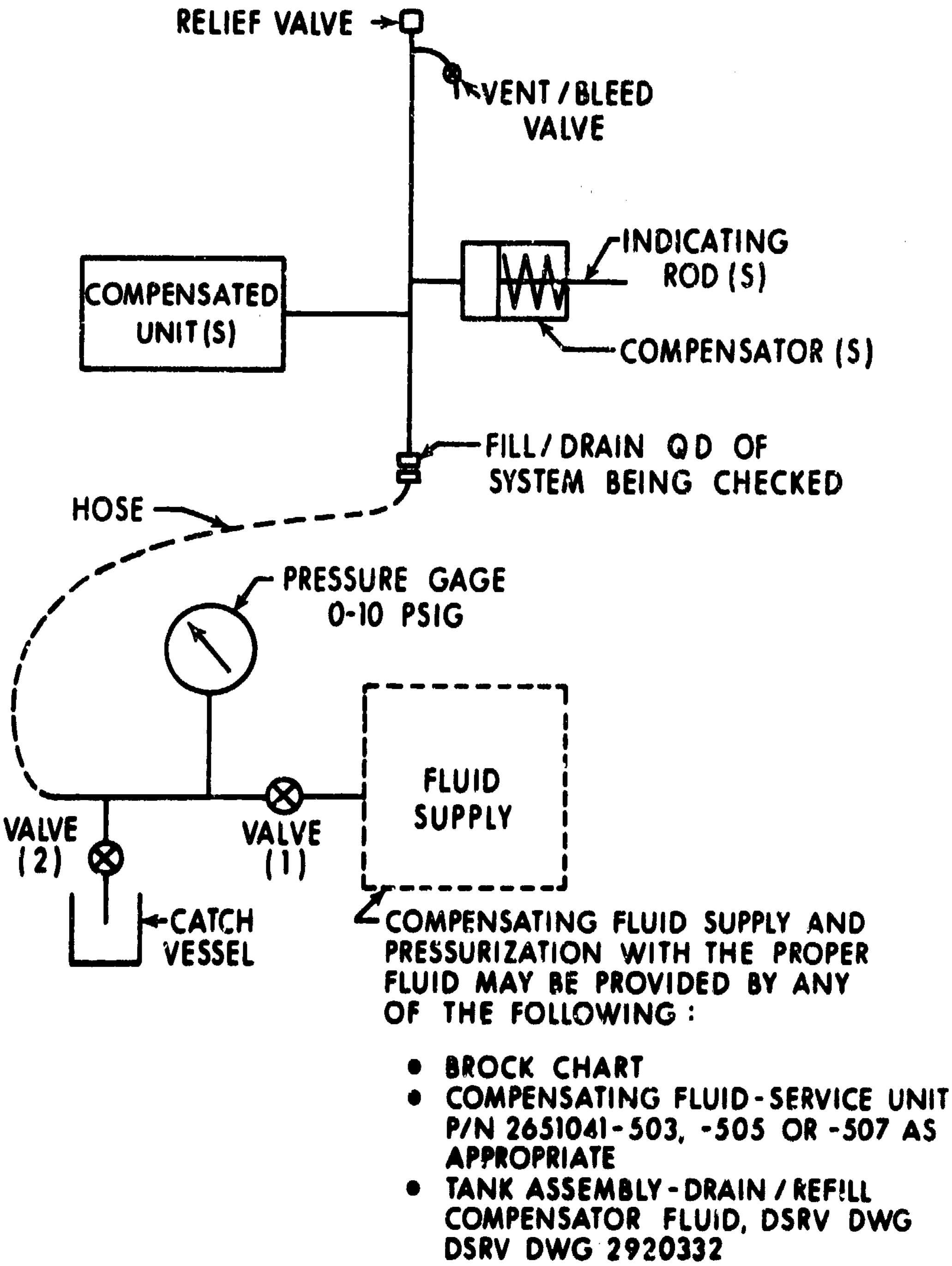

Figure 1 - Air Volume Check - Equipment Set-Up

Enclosure (8)

DSRV COMPENSATED SYSTEMS
Volumes

System	Total VOL Inc. Comp. (In^3)	Compensators Type	Compensators No.	Comp. Vol.	Ratio Comp/ Total	Proposed Air Limits (%)	Proposed Air Limits (In^3)
• Hydraulic System	8401.0	In pwr. units In H.D. winch	2	1012.5	.1210	0.25	16.8
• Trim/List System	Oil-13,100 Hg - 7,050 Tot.20,150	LMSC (Large)	2	1428.0	.0714	1.50	302.0
• Fwd Thrusters (Horiz. + Vert.)	572.3	LMSC (Small)	2	175.8	.3080	1.50	8.5
• Fwd J-Box	1195.2	LMSC (Small)	1	87.9	.0785	1.50	18.0
• Fwd Main Battery	9750.0	LMSC (Large)	3	2142.0	.2210	1.50	146.0
• Fwd P.D. Box	13,855.0	LMSC (Large)	2	1428.0	.1030	1.50	208.0
• Shore Power I/L Box	1969.0	LMSC (Small)	2	175.8	.0895	1.50	29.5
• Mid-Body J-Box	1217.2	LMSC (Small)	1	87.9	.0721	1.50	18.2
• Aft P.D. Box	13,855.0	LMSC (Large)	2	1428.0	.1030	1.50	208.0
• Aft Main Battery	9750.0	LMSC (Large)	3	2142.0	.2210	1.50	146.0
• Aft J-Box	1213.8	LMSC (Small)	1	87.9	.0725	1.50	18.2
• Aft Thrusters (Horiz. + Vert.)	572.3	LMSC (Small)	2	175.8	.3080	1.50	8.5
• Main Motor	744.1	LMSC (Small)	2	175.8	.2360	1.50	11.2
Totals							1138.9

5.4 GENERAL SURVEY INFORMATION

Excerpts from all compensating system surveys which were contributed, as per the original survey outline.

II Compensating Device

B. Materials selection for compensated system compensator housing or case material. Bare metal exterior or protective coating, paint?

Type	Material, Coating
1. Distribution boxes, hydraulic system, and other box-type enclosures	• 316 or 316L* stainless steel (both passivated and unpassivated) • 6061-T6 aluminum; anodized per specification MIL-A-8625, type II, class 1. Aluminum surfaces which require a smooth surface for O-ring seals, etc are anodized per MIL-A-8625, type III, class 1. • steel, mild and corrosion-resistant; coated with Dement coat and painted with acrylic • PVC (weldable grades of this material are available); bare surface does not corrode
2. Cylindrical housings for spring-loaded piston, bellows, or rolling diaphragm compensators	• 304, 316, 316L stainless steel; passivated, unpassivated; bare or epoxy painted • 6061-T6, 5083, 5086-H32 aluminum alloys; bare, anodized, or painted with acrylic

*For welded construction, type 316L (low carbon) stainless steel is desirable to minimize carbide precipitation in the weld grain boundaries and thus provide good corrosion resistance.

	• naval brass
	• Monel, bare
	• PVC sch. 80 pipe section
3. Self-pressurizing hydraulic reservoir (exposed to sea water)	• 356-T6 aluminum, anodized per MIL-A-8625 and painted with polyurethane (Laminar X-500)
4. Bladder housing	• Plexiglas (acrylic plastic)
	• fiber glass (GRP)

Elastomer type, physical properties, and MIL-SPEC?, Supplier:

Type	Has been used with the following fluids:
Buna - N or Nitrile (most widely used material)	MIL-H-5606B, MIL-H-6083C, ATF with zinc, Houghton(TM) PR-1192, Brayco(TM) 762, Hoover(TM) Submersible #2, Silicone fluids (this use questionable since fluid exhibits a strong solvent characteristic and shrinkage of nitrile elastomers has been reported), MIL-L-6081C-1010, Kerosene, Markol(TM) 52
Neoprene	MIL-H-5606B, 6083C, mineral oil, Phillips(TM) Magna #150, ATF with zinc, Markol(TM) 52

If spring is used, what type, material, and is it exposed to seawater?

For compression springs which operate in sea water, the following materials have been used:

- 17-7PH stainless steel (stress corrosion resistant)

(TM)Trade names marked with this symbol are proprietary to the manufacturer.

- 304, 316 stainless; painted and bare
- Elgiloy
- Be Cu
- Monel, K-Monel per QQ-N-286, class A, age hardened
- CRES (corrosion resistant steels); bare or passivated per QQ-P-35, type II

Use of external screens, baffles, or filters to minimize damage from sea-water particle contamination on elastomer or piston and internal housing. Give details, mesh size, or filtration:

- Screen, NiCu, 10 X 10 mesh (for rolling diaphragm cavity protection).
- Expanded metal mesh for elastomer protection against falling objects and when vehicle is serviced.
- Teflon scrapers on moving cylindrical surfaces.

Marine fouling, corrosion protection:

(a) Paints and coatings, metal passivation

- Catalyst resins
- Various multi-coat marine epoxy paints
- Polyurethane paints. Ex. Laminar X-500
- Dry lubricant coating
- Flame-sprayed zinc coating: galvanizing
- Thio(TM)-Tuck (polysulfide sealant)
- Flexane(TM) (No. 95 putty or liquid)
- Passivation of steels
- Various specification anodizing processes for aluminum and aluminum alloys; Sanford Hardcoat or Martin Process: 0.002 in. to 0.006 in. thick aluminum oxide coating; per MIL-A-8625

(b) Galvanic protection

- Zinc blocks as sacrificial anodes

(c) Use of protective greases, lubricants, bedding compounds

Name or Type	Vendor/Manufacturer
• Corrosion preventive compound, solvent cut-back, cold-application MIL-C-16173, grade 1	
• Dry film lubricants	Everlube Corp. of America, North Hollywood, Calif.
• Loctite™ Thread lubricants, sealants, and bedding compounds	Loctite Corp. Newington, Conn.
• Silicone grease (electrical grade) Dow Corning™ #11, #33 and #DS-4	Dow Corning Corp. Midland, Mich.
• Mystery™ Oil (high-penetrant lubricant)	
• Caulk-Tex™ (epoxy cauking/bedding compound)	Travaco Laboratories Chelsea, Mass.
• Lubri-Plate™ Grease #2, Marine A and #130AA, L-702 for close-fitting surfaces, threaded components; to impregnate protective felt pads for rotary shaft seals; O-ring installation; sleeve bearings and lip seal applications; for sliding shafts.	Fiske Brothers Refining Co. Newark, N.J.
• Silicone Compound MIL-S-8660	
• Silicone O-ring grease	
• Permatex™ 51, Grade 2	Permatex Co., Inc. Brooklyn, N.Y.

- Various specification greases which exhibit good water washout resistance:

  (1) MIL-G-81322 (supersedes MIL-G-7711), grease, aircraft, general purpose. Low torque resistance at low temperatures.

PED(TM) 3040	Standard Oil Co., of California
TG(TM) 4719 Regal AFB-2	Texaco Inc.
Aeroshell(TM) 6, Shell SG6127	Shell Oil Co.

  (2) MIL-G-24139 (NAVY); similar to previous specification, but lighter greases for ball and roller bearings, NLGI No. 1-2.

Mobil(TM) 28	Mobil Oil Co.
Shell(TM) Multi-Purpose Quiet Service; Aeroshell(TM) 22 (provides some rust inhibition)	Shell Oil Co.

  (3) MIL-G-23549, grease, general purpose, NLGI No. 2-3 and requires higher torque, good bedding lubricant. Aircraft catapult foot-pad grease.

PED(TM) 4147	Standard Oil Co., of California
AMOCO(TM) Supermil Grease No. 94532; Gov't designation LG0878	American Oil Co.

ROYCO(TM) 49B; Gov't designation C728 — Royal Lubricants Co.

IV. Special Design Requirements

A. Use of protective devices in compensated system. Enclose system diagram, if possible.

1. Sea-water leak detectors. Description, location

Direct survey excerpts:

- CRES electrode approximately 1/8-inch above lowest point of enclosure. Alarm at approximately 10^3 ohms from electrode to enclosure or other electrode. Used only in enclosures containing electrical equipment. Drain or sampling plug generally located close to electrode location.

- Small copper plates, 1/8-inch separation, mounted in an insulated block low in compartment. Sense leakage resistance between plates.

- A sea-water detector is located at the low point of each compensated container or return line for hydraulic reservoir. The detector consists of two brass plates of approximately 1.0 in.^{2} area. Plate separation is 0.35 inch and voltage across the plates is 28 vdc. With electronic amplification, the device can detect a 2% (approx) sea-water emulsion.

- Two phosphor-bronze electrodes spaced 1/4 inch apart and located 1/8 inch above bottom of motor casing. Conductivity of sea water used to actuate visual indicator.

2. Vent or relief valves

(a) Fluid expansion or electrical equipment gassing relief

Valve configuration, location, relief pressure

Pop-off relief valves are generally used at the top of the equipment enclosure, compensating piping, or compensating device.

Compensator	Pressure Setting, psid
Bladder (fully constrained by contoured housing)	10
Bladder	2-6
Piston, spring-loaded	2-5
Rolling Diaphragm	5-8 2-4
Self-pressurizing or bootstrap hydraulic reservoir	70 psid (operating pressure 50 psid)

3. Fluid volume indicators or device to monitor compensator volume:

Visual:

- Inspection as to fullness of elastomeric device.
- Rod projection (of piston, rolling diaphragm devices).

Remote Indicating:

- LVDT
- Magnetic reed switch (hard-shelled) and actuating magnet rides on reservoir piston; indicates when reservoirs are at low level (1/4 full).

C. Sea water to compensating fluid seals in system.

1. Type, military specification No. (if known), material, where used, estimated leakage rate.

(a) Static seals

Flat gasket or O-ring seals of Buna-N (Nitrile) material are generally used.

(b) Rotating seals

- Face seal fluid leakage rate reported in two applications was approximately 30 cc/hr. One rate quoted was for a carbon ring face seal operating at 900-1800 rpm.

- For TV camera pan-tilt rotating seals, a fluid leakage rate of 1 pint to 1 quart per day was reported. An inner tube compensator was mounted in a position physically below the level of the TV camera to provide a slight positive pressure (hence positive leakage) and prevent sea-water intrusion.

V. Compensating System Operation, Maintenance

F. Drain and flush procedures

Flushing with deaerated fluid was found to be advisable. For MIL-L-6081C, a kerosene flush was effective.

G. Fluid reconditioning process, if any

To "dry" compensating fluids, one contributor has used silica gel.

H. For compensator overhaul, describe cleaning and reconditioning of metal and elastomeric parts - what cleaning solutions or solvents are used?

- High-pressure kerosene spray or trichlorethylene for MIL-L-6081C-1010.

- Solvent, specification MIL-C-81302, Type 1

- Soap, specification P-S-600

- Fresh water

- Freon TF solvent

  Note: Solvents which contain chloronated hydrocarbons will damage electrical varnishes in motors or other portions of insulation systems, thus extreme care must be taken.

- B3240(TM) Aluminum Oxidation and Corrosion Remover; Arnett Assoc., Gardena, Calif.

- Manganesed(TM) Phospholene #MP-7; Bellflower Chemical Manufacturing Corp., Bellflower, Calif.

I. Compensator test methods, if any

- A 5 psi leak check is conducted at periodic maintenance intervals. Acceptable leakage rate: zero.

- See DSRV-I survey enclosure (6), page 7, No. 5

CHAPTER VI

GLOSSARY OF TERMS

Accumulator - A container device in which fluid is stored under pressure as a source of fluid power.

Actuator - A device which converts fluid energy into mechanical motion.

Bedding Compound - A thick lubricant or resilient, puttylike material which is used to protectively coat mating or adjoining surfaces.

Bleed - To remove or vent pressurized fluid and/or air.

Bulk Modulus - A measure of the resistance of a fluid to compressibility; usually defined as the reciprocal of compressibility.

Cavitation - A localized gaseous condition within a fluid stream which occurs where the pressure is reduced to the vapor pressure of the fluid.

Check Valve - A device which permits fluid flow in one direction only.

Circuit - A complete path of flow in a fluid system.

Component - A single element or part.

Depth Filter - A filter containing porous materials which primarily retain contaminants within a tortuous path.

Contaminant - Detrimental liquid or particulate matter in a fluid.

DOT - Deep Ocean Technology (Program).

Emulsifying - That property of a fluid which enables it to encapsulate water contamination so that very small droplets are maintained in a stable suspension.

Enclosure - An equipment envelope, housing, or case.

Fail-Dead	- A failure mode which precludes even partial operability.
Fail-Sick	- A failure mode which permits some degree of operability.
Fluid	- As used in this handbook, includes all candidate compensating liquids with the exception of sea water or fresh water.
Inhibitor	- A fluid additive which prevents or slows such chemical reactions as corrosion or oxidation.
Laminar Flow	- A streamline flow condition in which fluid particles move in continuous parallel paths.
Line	- A tube, pipe, or hose which acts as a conductor of fluid.
Linear Actuator	- A device for converting fluid energy into linear motion, such as a hydraulic cylinder.
LVDT	- Linear variable displacement transformer; an electrical device which measures linear displacement.
Manipulator	- A remotely controlled, armlike device which can locate and orient a tool or terminal device to perform a task.
Micron	- Approximately 0.00004 inch or one-millionth of a meter.
Mixing Leakage	- The local transfer of fluid and contaminants across a dynamic seal at liquid-liquid interfaces.
Nonemulsifying	- That property of a fluid which resists the formation of a stable suspension with water contamination.
Open Center	- A neutral valving position whereby pump delivery recirculates freely to sump.
Port	- An opening at a surface of a component for the intake or exhaust of a fluid; the external or internal terminus of a passage in a component.

Pressure Differential	- The pressure difference across a preselected point of reference.
Proof Pressure	- A nondestructive test pressure which is in excess of the maximum rated operating pressure.
Reservoir	- A sump or container in which fluid is stored in a fluid power system.
Seal Clearance	- The interfacial clearance between a given dynamic seal and the shaft or bore which is being sealed.
Servo (Servomechanism)	- A feedback control mechanism which, when subjected to the action of a controlling device, will operate as if it were directly actuated by the controlling device. A servo is capable of supplying power output many times that of the controlling device, this power being derived from an external and independent source.
Servo Valve	- A valve which controls the direction and quantity of fluid flow in proportion to an input signal; a slave or follow valve.
Specific Gravity (liquid)	- The ratio of the weight of a given volume of liquid to the weight of an equal volume of water.
Spool	- A term which has been loosely applied to almost any moving cylindrically shaped part of a hydraulic component which moves to direct flow through the component.
Surge	- A transient rise of fluid pressure in a circuit.
Turbulent Flow	- A condition wherein fluid particles move in random paths; the velocity at a given point varies erratically in magnitude and direction.
Vapor Pressure (liquid)	- The ambient pressure at a given temperature under which a liquid is in equilibrium with its own vapor.
Vent	- An outlet port for the escape of a gas or liquid or for the relief of pressure; the act of relieving.

Viscoelastic	- That non-Newtonian property of a fluid whereby viscosity is dependent upon the rate of fluid shear. Fluids such as MIL-H-5606B and MIL-H-6083C exhibit a large elastic component of viscosity.
Viscosity	- A measure of the internal friction or resistance of a fluid to flow and is the evidence of cohesion between fluid particles.
Viscosity, Absolute	- The ratio of shearing stress to the shear rate of a fluid, usually expressed in centipoise.
Viscosity, Kinematic	- The absolute viscosity divided by the fluid density, usually expressed in centistokes (cs).

CHAPTER VII

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1. It is suggested that the handbook could be improved by making the following changes:

2. Reasons for suggested changes are:

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13. ABSTRACT

Candidate approaches to depth/pressure-compensating devices and systems for fluid-filled deep submergence equipment are described and analyzed. Current design philosophies and considerations are discussed as well as the physical arrangement of compensating systems. A compensator design scheme and maintenance information have been included to provide guidelines and enumerate system design parameters and ramifications. It is planned that the handbook contents will be periodically revised and updated as 'feedback' information becomes available.

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